MACHINERY.

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AN INTERVIEW WITH CHIEF ENGINEER MILLIGAN

OF THE BATTLESHIP "OREGON"—INTERESTING FACTS ABOUT THE ENGINEERS AND MACHINERY OF THIS FAMOUS VESSEL.—INCIDENTS OF THE RUN AROUND CAPE HORN AND OF THE FIGHT AT SANTIAGO.

F. F. Hemenway.

Reference has previously been made in these columns to the altogether remarkable run of the battleship "Oregon" from Puget Sound to Jupiter Inlet, her immediate entrance upon duty off Santiago, and the subsequent glorious part she took in the running fight that terminated in the sinking of the grandest fleet Spain could put afloat under the command of her most gallant and chivalrous Admiral Cervera, who fought with desperate energy and courage against the enormous odds of brains in the engineering department of our battleships, no less than brains behind the guns and brains on the bridge. If the great unprofessional majority of the country has been slow in discovering and appreciating the vast contributive importance of the engineering department of our new navy when the question comes to be one of winning naval victories, that lay contingent is opening its eyes under the pressure of hard facts, and when the "great public" sets itself earnestly to the task of thinking, it may be depended upon to think right. Without detracting in the least from the glory of the officers and men of the navy, the men who fight in full sight of the enemy, the public is fairly awake to the fact that to fight at the guns when the great fighting monster is where there is opportunity to fight is one thing, and to put her in that position through 15,000 miles of intervening watery space, all her machinery of propulsion and of fighting in perfect condition, is equally another thing.

Asst. Eng. W. D. Leahy. Asst. Eng. J. M. Reeves. Asst. Eng. Frank Lyon. Chief Eng. Robert W. Milligan.

ENGINEERS OF THE "OREGON."

Machinery, through the courtesy of Chief Engineer Milligan, is enabled in this issue to place before its readers a photograph of himself and three of his assistants—W. D. Leahy, J. M. Reaves and Frank Lyon. The photograph was taken on shipboard, in what we should call workaday attire. The picture speaks for itself.

The "Oregon" left Puget Sound March 6, and arrived at Jupiter Inlet May 24, as the sequel proved, as ready for what she was built for as when she left for her sail over the two great oceans and through the straits. The long run of 14,510.9 knots was, Chief Milligan said, so far as engineering matters went, without incident of consequence, no stop being made on account of mechanical derangement. Such stops as were made were for coal or for purposes in which the engineers' department had no especial concern.

The total amount of coal burned was 3,908.84 tons, which, looked at in the aggregate, appears to be a big quantity of potential energy unloosed, but when the effect is scattered along a line representing a distance equal to more than that of half way around the earth, looks small considering the number of tons of water that were displaced along that line.

The steaming by distance between stopping places varied from a trifle more than 10 to 1.46 knots per hour, one run under forced draught being at the last named average speed.

In charge of Chief Milligan at the present time are 98 men, which is one short of a peace complement. This force consists of one passed assistant engineer, two assistant engineers, two naval cadets (engineer division), four chief machinists and four each first and second class machinists; the balance of the force consists of eight oilers, six water tenders, first and second class firemen, coal passers and one yeoman. The high grade machinists perform engineers' duties. These are the men who feed the hungry furnaces, getting steam through a labyrinth of pipes to a little multitude of big and small cylinders that are always waiting; the men who set free the pent-up energy of the coal bunkers and guide it in the mechanical work of propulsion and fighting; the men who brought the great pacificator from her distant anchorage; brought her home on time and in condition, and put and kept her in position for her work, her intricate fighting machinery supplied with the necessary potentiality for handling the enormous atoms of destruction that decide battles at sea.

Chief Milligan spoke in high terms of the work done by the Union Iron Works, almost literally nothing being done to the engines and machinery during the run, a few tubes requiring tightening and a slipper for some cross-head being renewed, matters of trifling consequence, he said, which, being interpreted in the light of Chief Milligan's altogether evident intention to hold up all the responsibility he can find, and to give everybody else, including his subordinates, something greater than exact justice, signifies that the engines and machinery were so well built and erected, that with the aid of his trained engineers and mechanicians, he was able, through the exercise of great care and skill, to bring the "Oregon" through her long run in just about absolutely perfect condition to go to her position off Santiago, where there was no such thing as anchoring—simply drift a little and steam a little. This was not exactly what Mr. Milligan said, but it is what every reader of MACHINERY who is an engineer or mechanic knows to be true.

Among other things he seemed to be particularly pleased that he had no trouble with condenser tubes. Those acquainted with the use of surface condensers are aware that the intention to use tubes so thin, in order to get the greatest possible effect from a given exposed surface, has gone so far as to outrun the ability to pack such tubes so that they shall remain tight; and this has caused serious trouble not only in the navies of the world, but in the merchant marine service. A vessel like the "Oregon," 2,000 miles from the start, with more than six times that distance to go, and an almost impenetrable network of leaky condenser tubes, is not pleasurable either in contemplation or realization. It is a magnificent nightmare, with Olympus piled on to aggravate the congestion. Chief Milligan was understood to say that the builders of the "Oregon," at their own suggestion and expense, substituted tubes two numbers thicker than called for in the specification. To this he attributes the absence of any trouble from leaky and failing condenser tubes. He believes that the making of them so extremely thin has been carried too far—that the limit has been exceeded—when tubes so thin are employed that the matter of leaks is always a present and a pertinent quantity. He not only had no trouble with leaky tubes, but he maintained a vacumm of 28 inches, in both of which respects, considering the time under steam, the record is more than remarkable.

There was no trouble with leaky flanges of steam pipes. Any ashore engineer, who from observation has made a mental estimate of the number of flange joints—to count them would be impracticable—around the machinery of a vessel like the "Oregon," joints in steam pipes that seem to wander here and there and pretty much everywhere for purposes almost beyond fathoming, will certainly wonder that these pipes and joints came through such a run sound and tight. That there were no leaks worth speaking about anywhere is made apparent by the fact that the "Oregon" arrived at Jupiter Inlet with clean boilers full of water containing scarcely more than a trace of salt—something that would be remarkable in a run of one-quarter the distance traversed by this ship.

The flanges on all these pipes, large and small, are so heavy that many would characterize them as clumsy, which Chief Engineer Milligan rather shrewdly concludes is one of their strong points; they are heavy enough to hold their packing, which, though not the chief end of a flange joint, is a very essential accompaniment. The flanges, he said, were not only heavy, but every pair came together perfectly fair. Every engineer afloat or ashore knows the difference between pipes fitted in this way and those fitted so that there is a constant spring leverage to pry flanges open at one side or the other, making the blowing out of the packing a question of a short time under the present day practice as to steam pressure.

When asked how he packed all these joints to insure such continued tightness, Chief Milligan disclaimed all credit in the matter, saying that such flanges, fitted as they were, could be packed tight with any sort of ordinary good flange packing. The question, he said, was asked him by engineers of foreign battleships, and about the only reply he could make was that the joints gave him no trouble whatever, and that he should not hesitate to use anything for packing that had reasonable fitness for the purpose.

The writer, who has had about the usual experience of a landsman with such joints, and would rather cheerfully admit, a priori, a feeling of positive discouragement at the prospect of keeping such a multitude of joints tight for such a time, tried to pry further into the matter, and was told that unquestionably the extended experience of the Union Iron Works in the construction and erection of hydraulic mining machinery, where high pressures of an obstinate fluid had to be dealt with under conditions generally the reverse of favorable, had a good deal to do with the eminently successful piping from the "Oregon's" boilers.

Referring to the decisive running fight at Santiago, a fight that apparently settled the war, when the Spanish fleet initiated the fight that was so short and withal decisive, Chief Engineer Milligan's force and equipments were ready for the emergent action required. He had expressed himself forcibly in favor of keeping steam on all the boilers, arguing that it was contrary to all reason to put immense boiler capacity into a battleship and have part of it in a condition little worse than that of uselessness when the emergency came, and though furthest away when the action began, the "Oregon" soon took her place next the "Brooklyn" and held it to the end. Considering the disadvantages of a foul bottom, the fact that she steamed during the engagement practically at her trial speed, when she was clean and polished, as it were, and used picked coal, makes comment as to the condition in which her machinery was kept something more than superfluous.

The regulation in regard to forced draught was disregarded—forgotten is a better word—a pressure of 3 inches being maintained instead of the regulation 1½ inches. The experience during the long run from Puget Sound was favorable to the engineer force in regard to this, as at least on three different occasions forced draught was resorted to, giving them the experience that is generally wanting.

Asked about the sensation below when the big guns were pounding away just overhead, Chief Milligan admitted that one probably would notice it if he had nothing more serious to do. The temperature, he thought, might have been a little high at times. The noise of the guns was rather inspiring than otherwise. At one time, when the chase was for the "Colon" and the guns were silent, his assistants appealed to him to use his influence with Captain Clark to give an occasional shot at random, just to make things appear lively, and Captain Clark is authority for the statement that the appeal was made. Everything that transpires overhead during a fight is at once communicated to the engine room, so that those who are below are kept informed of the progress of things overhead.

Chief Engineer Milligan is a noticeably intelligent-looking man, as are his chief assistants. He is about fifty-five years of age. He was in the engineering branch of the service during the civil war, remaining in active service until about 1870, when he entered the Naval Academy, remaining for three years. Since then he has served as instructor at the Naval Academy, and for a time was stationed at the Mare Island Navy Yard. He was a member of the Board of Inspection and Repairs for a time, after which he was appointed to his present position, where he has made an unquestionable record. Quiet and unassuming, he is deservedly popular with those with whom he comes in contact. When spoken to in regard to the responsibility he must have felt, and the work he must have done in the engineering work of bringing the "Oregon" from her distant anchorage at a time when so much was expected of her, he turned the matter aside by the altogether characteristic remark that by their ability and promptness and entire willingness, his assistants released him of the greater part of both responsibility and work. Captain Clark says that during the chase after the "Colon" he found him most vigorously shoveling coal into one of the furnaces.

Chief Milligan made one remark concerning the fight at Santiago pertinent to a good deal of recent discussion concerning woodwork on war vessels. When he was a member of the Board of Inspection and Repairs there was one member whom they good-naturedly voted a crank for his persistency in advocating the abolishment of wood everywhere. After the fight he said he took the first opportunity to write to this member that he (the member) was wholly in the right. The Spanish vessels, set on fire by our shells, reminded him of nothing so much as a very considerable conflagration in a city. Pointing to a chest of drawers and other articles of furniture in his room, these, he said, must all be replaced by iron or steel furniture.

AMONG THE SHOPS.

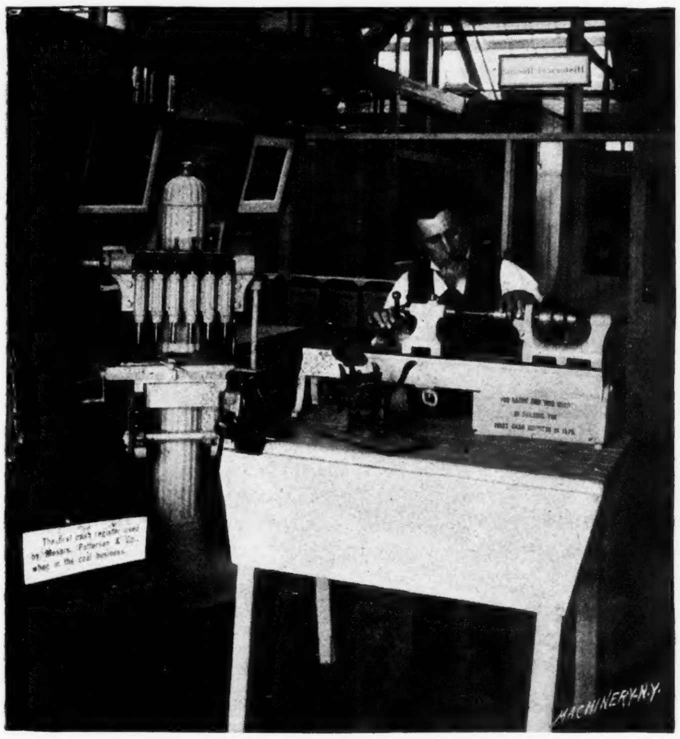
MORE NOTES FROM THE NATIONAL CASH REGISTER COMPANY'S WORKS.

For the benefit of new subscribers who may not have seen my notes in the September issue, relating to the factory system in vogue at the National Cash Register Co., of Dayton, O., I will say that this system has attracted the attention of so many writers that it has become widely known as "The Dayton Plan," and is probably familiar to most of them. The fundamental principle of the whole plan seems to be to secure the intelligent co-operation of the employees through humane treatment and the offer of prizes to stimulate individuals and departments to their best efforts, at the same time permitting them to govern themselves by a system of committees instead of working under a superintendent or general manager. Among the prizes offered is $600 to be divided every six months among the fifty employees who submit the best suggestions for improvements in mechanical or business matters; thus every one has a market for his ideas and a chance to display his ability.

SHOP SYSTEMS.

The ideal of the N. C. R. Co. is high, and from a glance through the tool supply room it is evident that they strive to maintain it in mechanical as well as other matters.

The jigs and tools are remarkable for their finish and perfect condition, and it would certainly be more satisfactory to most toolmakers to be permitted to finish up their work in such a



THE FIRST CASH REGISTER AND THE LATHE USED IN BUILDING IT IN 1879.

manner. The absence of dirt, dull tools, toothless milling cutters, etc., was conspicuous. The number of tools designated by a poster at the entrance of the tool supply room was 29,202. I did not make an individual examination to prove that the same condition existed in all cases.

To aid in keeping track of such a number of tools they are carefully marked and a double check system adopted with gratifying results. The usual round tool checks are supplemented by checks of a rectangular form. To begin with, there is a typewritten list of all tool check numbers in regular order posted in the tool supply room, and on the same line with the number of the check follows the holder's name.

On another typewritten list is the name of every tool alphabetically arranged, one line across the page, reading thus:

"No. 36A.—8R. Key cam gauge for back cut on No. 8——2d"

This is from the section of the list under the letter K, and the tool would be stamped in exactly the same manner to insure its identification and return to its proper place. The receptacles for tools are arranged in sections and lettered; 36A—8R means that the key cam gauge is kept in the 8th perpendicular row from the end, and on the 36th horizontal tier of section A, and the second d at the end of the line shows there is more than one division in the pigeon hole, and that the tool is kept in the second one from the front. In this particular case there was a drawer with three divisions in the pigeon hole. In the section with the tool is a pin for the workman's check, and at the end of the tier is another pin with some of the rectangular checks marked A—36. Now, we will suppose that the person who is at work milling the back cut on No. 8 key cams for No. 35 ma-

THE N. C. R. CO.—TOOL SUPPLY ROOM.

For No. 35 Machine.	Tool Check.
Gauge for back cut on No. 8, Key cams........................	No.................
....	No..............33
....	No.................
....	No.................

Signed, PHILLIPS.

FIG. 7.

chine (every different style of machine is known by a number) holds his allotted number of tool checks (numbered, say, 33) and he wants the gauge. One of his tool checks would be sent to the tool supply room by the errand boy, with an order like Fig. 7.

If necessary the tool room attendant would refer to the typewritten list, which would tell him exactly where to find the tool wanted. The workman's check would be left on the pin in the place from which the tool was taken, and the gauge sent to the

Tools Approved by
No. 35 COMMITTEE.

VERBAL ORDERS DON'T GO.

FIG. 8.

workman with one of the rectangular checks marked A—36. From this it will be seen that he would always have the same number of checks on his ring, the rectangular ones showing how many tools he had out and the approximate location of their place in the tool room, and he can tell at once if he has lost a check.

Where men are leaving at short notice, as they frequently do in a large crew, it is a convenience to be able to take an employee's check ring and tell in five minutes exactly what tools

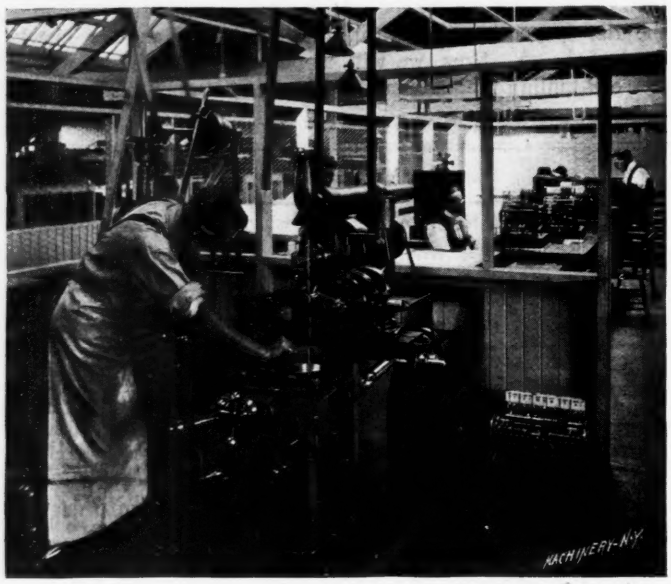
NO. 79 PRINCIPLE. TOOLS ORDERED FROM TOOL ROOM.

Name and Dimensions of Tool	Quantity.	Date of Order.	Order Number.	Date Finished.	Cost.
....	....	....	....	....	....
....	....	....	....	....	....
....	....	....	....	....	....

FIG. 9.

he has out, without hunting all over the tool room for a few stray checks. Some one may have borrowed one of the tools and he has forgotten about it; but by going to the tool room the attendant can take the rectangular check, go directly to the tier and section, and find out what tool his check is on. The row and division could also be shown on the rectangular check if desired, which would locate the place precisely. To me, it seems as if it would be desirable to have one of these rectangular

checks for each tool, showing its precise location, to be kept on the check pin beside it, to be given out, the same as they are now, instead of making one check do for any tool in a single tier. In that way the man would hold one of the rectangular checks as a receipt for every tool he had out, enabling him to identify the tools charged to him wherever they might be found in the shop, and leave the tool room attendants no excuse for one workman obtaining his check on a tool charged to a shopmate, except in the case of small taps, drills, etc., which cannot be distinguished one from another.



A SECTION OF THE EXPERIMENTAL DEPARTMENT.

As I have said, each style or model of register is known by a number; I understand, also, that they are classed under five general heads, as "No. 79 principle," (Fig. 9) or a similar term, according to the principles upon which they are constructed, the same as a distinction would be drawn in the machine shop between lathes, planers, milling machines, etc. Yellow, white, blue, red and green are the colors adopted to correspond to these five principles, and every stock order or any kind of a communication in the factory is written upon paper with a stripe of one color or the other across each end, as shown in Fig. 8. These stripes facilitate picking out a desired order or note from those on file, and is an employee's suggestion, of greater value than was at first apparent. Tote boxes, work tags, etc., are also distinguished by these colors.

Colors are utilized in the tool room for other purposes; also when a change is ordered on any tool, "Improvement ordered," with the date, is written upon a round red tag about the size of a dollar, and tacked in a conspicuous position beside the tool, and a reference to the record of improvements under that date explains it fully. "Don't give out" on a red tag is another form of wording for this danger signal, where temporary changes do not enable it to be used.

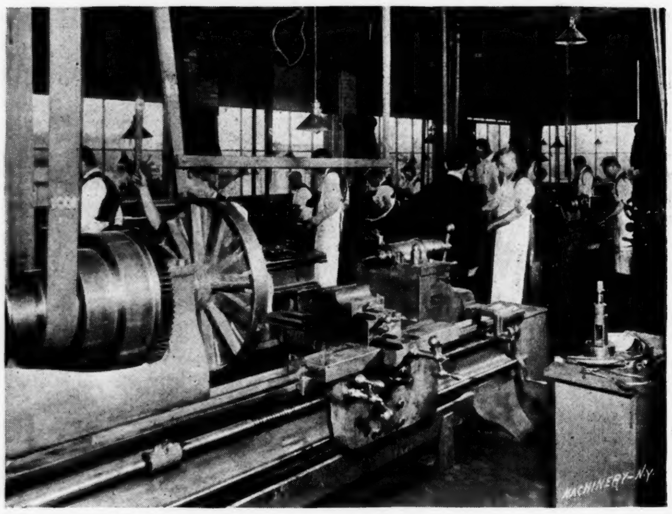
When a new tool is made, the various operations are performed by different men, the same as if building a machine, then put together and inspected by a capable all-round tool maker. When it has passed the final inspection, it is marked with the stamp of approval, thus, (A) and put in its place in the tool supply room, with a blue round tag tacked up beside it bearing the date. Unless the tool has been called for within a week, the attention of the foreman of the proper department is called to it, thus insuring his knowledge of its existence. When it has passed through service, and is pronounced all right, it is returned to its place and the tag taken down. Fig. 9 is the heading of a blank page from one of the books used in the tool making department, and needs no remarks, except that I believe there is one of them for each style of machine made.

As I passed through the shop I noticed a box of work near the aisle with a blue card attached, on which was printed "Move It." By inquiry I learned this was also the suggestion of an employee. Instead of hunting up the truckman or yelling across the shop at him, the workman puts such a card on the tote box when he is through with it, and no other directions are needed, because the truckman knows it is ready for him, and that every lot of work in his department goes to the inspector from each person who does any part of it.

The markings on the glass tops of the keys are made by an etching process, the glass being prepared in sheets coated with beeswax to hold a thin film of lead, which is rubbed down on to the glass while it is still quite warm. Girls stencil it and work out the opening in the lead by hand. The other side is also coated with lead, to protect it from the fluid, and it is immersed in a bath of hydrochloric acid, after which it is cleaned and the little disks cut out with a diamond attached to the end of a small vertical shaft, held in stationary bearings and turned by a hand crank. A circle showing through the glass coincides with the one described by the diamond, to make the location of the work quick and easy. There are five colors used for the ground work of these key tops, those for registering a cash sale being white, credit sales red, money paid out blue, received on account yellow and money changed (no sale) orange. The fact that the softer shades of yellow is the prevailing color in all parts of the factory suggests the joke that the machinery was all received on account.

The indicators, as they are called, which bob up before the customer when a key is pressed, are made of metal, with markings of gummed paper stuck on. Some of these bits of paper are purchased in large quantities, and can be put on quite rapidly by girls who are accustomed to it, but a miniature power press has been made for doing a part of this work. It was set up for the "Paid Out" indicator when I saw it, and doing very neat and satisfactory work. The strips of gummed paper were fed automatically, and the whole of the lettering punched through the dies and pressed firmly against the moistened surface of the indicator, the same stroke preparing the paper for the next indicator by removing the interiors of the O and A.

In the preceding letter, as it appeared in print, the expression



A CORNER OF THE TOOL MAKER'S DEPARTMENT, SHOWING LARGEST LATHE IN THE FACTORY.

"the entrance of the park" was used in such a way as to indicate that the section of the city surrounding the N. C. R. works is an inclosure, but such is very far from being the case. "Factory" is the word that should have been used instead of "park."

Limited space compels me to stop here, but I hope that some of the N. C. R. mechanics will be prompted to contribute something from the mass of exceptionally interesting shop notes which could be collected there. A. L. G.

* * *

It is stated by Theo. H. Curtis, in the "Railway Journal," that a steam gauge will read higher when hot than when cold—sometimes considerably higher. He says that this is a fact that can be tested by arranging the gauge so that it can be disconnected from the boiler, the water emptied out of the coil and returned to place. The water in the coil serves to keep the hot steam away from the gauge, and if a reading be taken before and after the removal, he claims that the latter will be higher. We should not advise trying the experiment, especially with a new gauge, as the presence of steam in the tube is likely to permanently injure the reading of the gauge.

WOODEN BACK CALIPERS.

OLIN SNOW.

Wooden back calipers are more of an instrument than their appearance indicates. With them it seems to be a case of "handsome is that handsome does." They are about as crude looking affairs as one can find, but as a shop measuring tool for work of large sizes, I believe they are not to be surpassed by any other form of instrument, however expensive or elaborate. It must be remembered that they are not intended to originate sizes, like a pair of micrometers or verniers. They are intended, simply, to test the diameter of a piece of work by comparing its length with some reference standard, like a reference gauge or the measuring points of a pair of verniers.

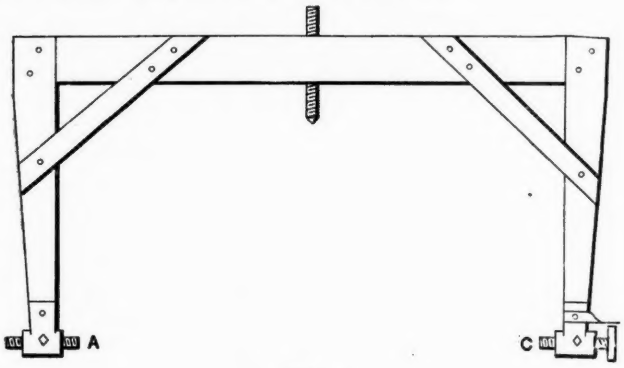


FIG. 1.

In Fig. 1 is shown one method of making wooden back calipers. They consist of a well-braced and a thoroughly seasoned wooden frame, which should be as light as is consistent with stiffness. At *A* and *C* are fastened two metal ends, which are chucked out and threaded with internal threads to receive the two-threaded measuring points. The point *A* can be adjusted either way and clamped firmly in position for slight differences in the diameter of the piece to be measured; but it is not advisable to allow for much adjustment, as it is more satisfactory to have several frames, each of which is of the right size for the work in hand. The point at *B* is made with an index wheel graduated to read to thousandths of an inch.

In measuring with wooden back calipers, they are first set to the exact size to which the work is to be turned, and the reading of the micrometer screw noted. Then, with the screw turned out far enough to allow the calipers to pass over the work it can be told at a glance just how much stock has to be removed. In Fig. 2 is shown how the calipers are supported in measuring. A supporting wire, *W*, is screwed into the upper beam of the calipers and is located midway betweeen the two measuring points. This wire has a knife edge at its lower end,

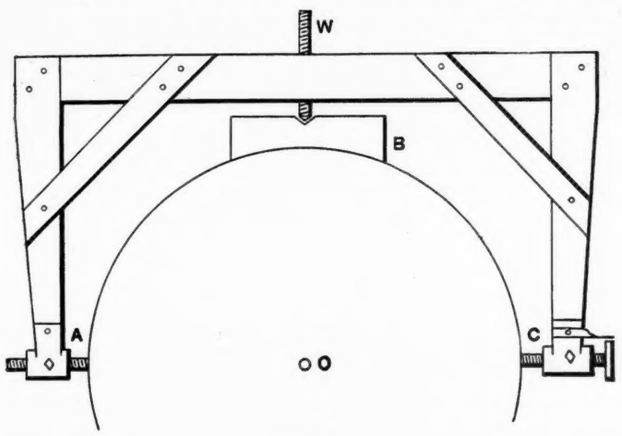


FIG. 2.

which rests in the notch in the upper side of a wooden block *B*, the lower side of which is sawn out to conform to the curvature of the work in the lathe. In preparing to measure, the wire *W* is screwed up or down until the measuring points come diametrically opposite each other. That is, the points *A*, *C*, and the center *O* of the work must be exactly in line. After this adjustment has once been made, it is very easy to keep the wire in adjustment, moving it down a little as the metal is turned off. Another thing to look out for is to have calipers in perfect balance when they are supported by the wire *W*. When thus adjusted it will be seen that they make a very delicate and easily manipulated measuring instrument.

One important feature, however, remains to be mentioned. It is, that by the use of these calipers the liability of getting the work below size is reduced to a minimum. If they are set so that the points *A* and *C* come diametrically opposite, they will, of course, measure exactly; if they are set either above or below the center, however. they will measure smaller than the actual size of the work, so the workmen need have no uneasiness about the setting of his calipers after they have once been set to the reference rod. If there is any error, it will be on the safe side.

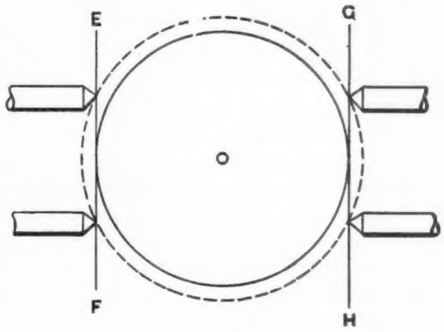


FIG. 3.

This is clearly indicated in Fig. 3, where the distance between the vertical lines *EF* and *GH* is equal to the correct diameter for the work. When the points are set to this diameter, however, and are above or below the center, as clearly shown, the work must be of a diameter corresponding to that indicated by the dotted circle, which is larger than the correct size.

Wooden back calipers are essentially a testing instrument for fine work. Before the finishing cut is taken it may be desirable

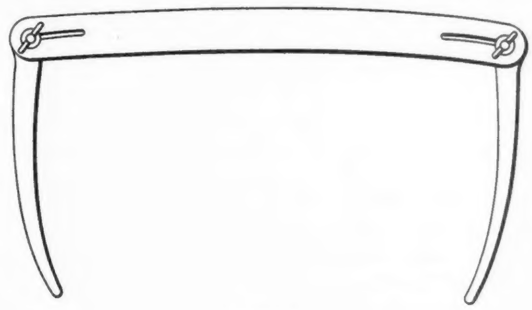


FIG. 4.

to do the measuring with another style of "wooden backs," shown in Fig. 3, which consists of a wooden bar for the back and two metal legs. These calipers are used like any ordinary pair of calipers, and it may be that this form is what is most commonly known to the readers as "wooden back calipers."

* * *

COURSE IN NAVAL ARCHITECTURE FOR ANNAPOLIS CADETS.

For several years past a course in naval architecture has been given at the Massachusetts Institute of Technology, Boston, Mass., under the direction of Prof. C. H. Peabody. This course has been extended and improved from year to year until, according to the New York "Sun," it has received very complimentary recognition from the Navy Department, through directions given by Secretary Long that the course in naval architecture begun at Annapolis last year be transferred to the Massachusetts Institute of Technology.

Constructor Hobson was the first director of the Annapolis course, and its abandonment is on the recommendation of Chief Constructor Hichborn, who was opposed to its establishment there at the time Hobson was first placed in charge one year ago. Commodore Hichborn did not believe it practicable for the students to receive the advantages or the attention from one instructor that they should have, and since Hobson and three cadets were detached last spring and sent to join the fleet, he has been examining the course at leading scientific schools of the country with a view to selecting one for Annapolis graduates. Eight cadets are to report this month for the course, which will cover a period of three years, including practical instructions during the summer at ship yards and navy yards.

This class will be the largest undergoing instruction for appointments to the Construction Corps chosen since this advanced course has been given to cadets.

FORMULAS FOR THE STRENGTH OF GEAR TEETH.

J. J. Flather.

In considering the strength of gear teeth we shall in the first place neglect the actual shape of the tooth and assume it to be rectangular in every section, as shown in Fig. 1.

Further, for the sake of simplicity, it will be assumed that the load, P, acts on the outer circumference of the gear, and hence at the extremity of the tooth. If we consider the conditions under which the load is assumed to be applied in actual practice, we shall find that:

(1) A rough cast gear may have the whole load concentrated upon one corner of a tooth; or the gear itself may be well made, but may be out of alignment, due to springing of shaft, bad workmanship or other causes; in this case also the load may be concentrated upon the corner of one tooth.

(2) The gear may be well made, with accurately cut or cast teeth, well mounted upon heavy shafts having proper alignment; in which case the load may be distributed over the entire breadth of a single tooth.

In determining a formula, then, for the pitch of a gear tooth, we must know the kind of gears we have to deal with and what considerations affect their strength.

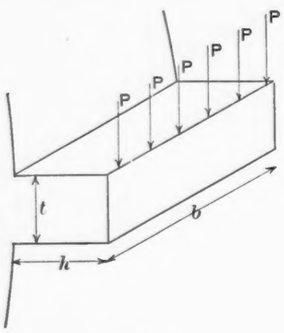


FIG. 1.

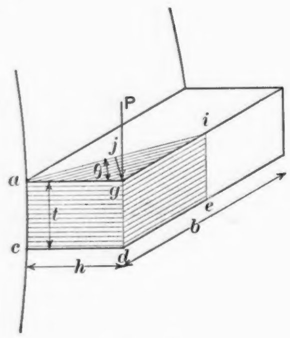


FIG. 2.

In the first case, if the gear is rough cast or placed so that the load may come upon one corner of a tooth, we must consider not only that the tooth is a cantilever, as in Fig. 1, but that the corner of the tooth is itself a cantilever, as in Fig. 2, in which the whole load, P, is concentrated at the point, g, so that the moment tending to break the corner off from the tooth is P multiplied by the perpendicular distance from P to the line, a i; hence, $P \times gj = Ph \sin \Theta =$ bending moment. Now the resistance which the tooth offers depends upon the character of the metal as well as its actual section, which in the present case is the rectangle a c e i. It can be shown that the resistance of this corner to breaking across is equal to

$$\frac{1}{6} f t^2 \times ai = \frac{1}{6} f t^2 \frac{h}{\cos \Theta} \quad (1)$$

in which f = stress in the metal. Equating this to the bending moment we have

$$P h \sin \Theta = \frac{1}{6} f t^2 \frac{h}{\cos \Theta} \quad (2)$$

or,

$$f = \frac{6 P \sin \Theta \cos \Theta}{t^2} \quad (3)$$

Noting that $\sin \Theta \cos \Theta = \frac{1}{2} \sin 2\Theta$, and substituting in (3) we have:

$$f = \frac{3P}{t^2} \sin 2\Theta \quad (4)$$

When $\Theta = 45°$, we have $\sin 2\Theta$ a maximum = 1; therefore

$$f = \frac{3P}{t^2}; \text{ or } P = f\frac{t^2}{3} \text{ and}$$

$$t = \sqrt{\frac{3P}{f}} \quad (5)$$

For new teeth, rough cast, in which $t = 0.48 p^1$ (p^1 being the circular pitch) we have:

$$0.48 p' = \sqrt{\frac{3P}{f}} \text{ or}$$

$$p' = 3.6 \sqrt{\frac{P}{f}} \quad (6)$$

For wooden cogs in which $t = 0.6 p^1$, we shall have:

$$p' = 2.9 \sqrt{\frac{P}{f}} \quad (7)$$

If in any case the width of face, b, is less than the height of tooth, the above formulae do not apply. In this case we have used the thickness of tooth without considering loss of strength, due to wear; but we have also assumed the whole load as concentrated upon the corner, whereas, as wear occurs, although the tooth itself is reduced in section, yet the load is better distributed and the tooth may be actually stronger.

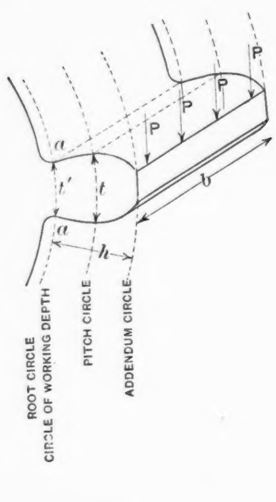


FIG. 3.

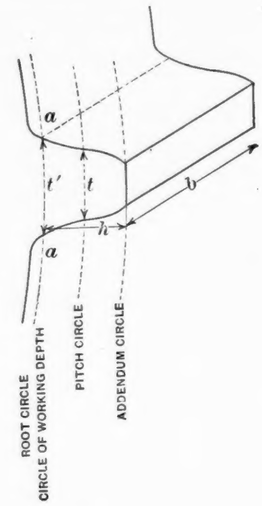


FIG 4.

In the case of rough cast iron and mortise gears in which the whole load may come upon the corner of one tooth, the width of gear does not affect its strength, and there is no advantage, as far as strength is concerned, in making such a gear wider than $h \times \frac{1}{\cos 45°}$, or 1.41 h; as h, the height of the tooth, is frequently made equal to $.7 p^1$, we have the safe width of tooth. $b = 1.41 \times .7 p^1 = p^1$ very nearly.

In order to prevent excessive wear such gearing is frequently made so that its breadth of face, $b, = 2 p^1$. So far we have assumed that the tooth section was a rectangle, but in practice the tooth may have the shape shown in Fig. 3, or that in Fig. 4, which represent a wide range in the strength of two gears, when accurately made and aligned, but for a load concentrated upon one corner the difference is not so marked as the shape of the teeth would indicate.

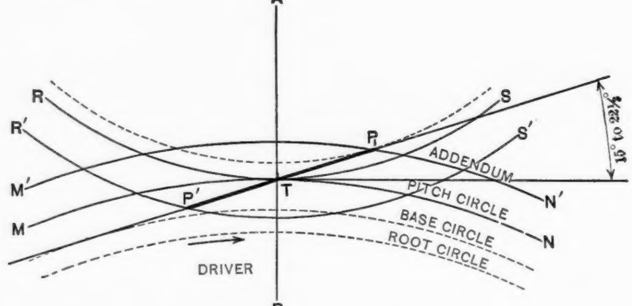


FIG. 5.

Investigation shows that if the smaller of two gears has at least 18 teeth and not more than 50, the strength of the tooth will be practically the same as that determined by the above formulas. Since these values may be said to represent fairly well the limits of the number of teeth in the smaller of two gears in those cases where rough gearing is permissible, we shall not at the present time discuss the influence of shape upon the strength of this kind of gearing.

Of more importance is the consideration of that case in which the load is supposed to be equally distributed between two pairs of teeth, the influence of which we believe to be very much overestimated. As this matter relates to the strength of all gear teeth we shall discuss its effects in general.

The point of contact between tooth surfaces in exact gearing can readily be determined when the system upon which the gears

have been designed is known. Thus in the involute system, the locus of the point of contact is the straight line passing through the line of centres at the point of intersection of the pitch circles and making an angle with the horizontal of 15° to 22½°, as shown by P T P in Fig. 5.

On the other hand, the point of contact between two tooth surfaces of the cycloidal system will always be upon a reverse curve, which is a portion of the describing circle for the faces and flanks of the tooth; thus, in the interchangeable system of 12-tooth base, i. e., in that system where the smallest pair of gears in the set is assumed to have 12 teeth and these have radial flanks, the diameter of rolling circle will evidently be equal to the radius of the 12-tooth pinion; and since the tooth profiles are generated by rolling this circle upon the pitch circles of any pairs of gears in the set, the point of contact will always fall upon the circumference of this describing circle when the centres of the latter lie upon the line joining the centres of the two gears, as shown in Fig. 6. If in Figs. 5 and 6 M^1 N^1 and R^1 S^1 represent the addendum circles, or the circles drawn through the tops of the teeth of the two gears, we shall have the two loci P^1 T P_1 of the points of contact, as shown in heavy lines; that is, contact will begin in each case at P^1 and, as the teeth move around, the point of contact between the surfaces will travel along the line or reverse curve, P^1 T P_1, until contact ceases at P_1. Now considering Fig. 6, if the pitch were equal to the arc, P^1 T P_1, one tooth would just be beginning action at P^1, while another would be quitting at P_1. In practice, in order to obtain smoothness of action the effective height of tooth is usually taken at about six-tenths of the circular pitch in cast gears, and about $\frac{64}{100}$ of the circular pitch in cut gears. If p^1 = circular pitch, p = diametral pitch, and h = effective height of tooth, we have: $h = 0.6\,p^1$ for cast gears; and

$$h = 0.64\,p^1 = \frac{2}{p} \text{ for cut gears.}$$

We would here note that the effective height of tooth is not the total height, as an additional amount, frequently $1/10\,p^1$ for cast gears, is allowed for clearance between the root circle and the tops of the teeth of the mating gear. With an addendum equal to half the above height of tooth it will be found that the average arc of contact for gears between 12 teeth and a rack,

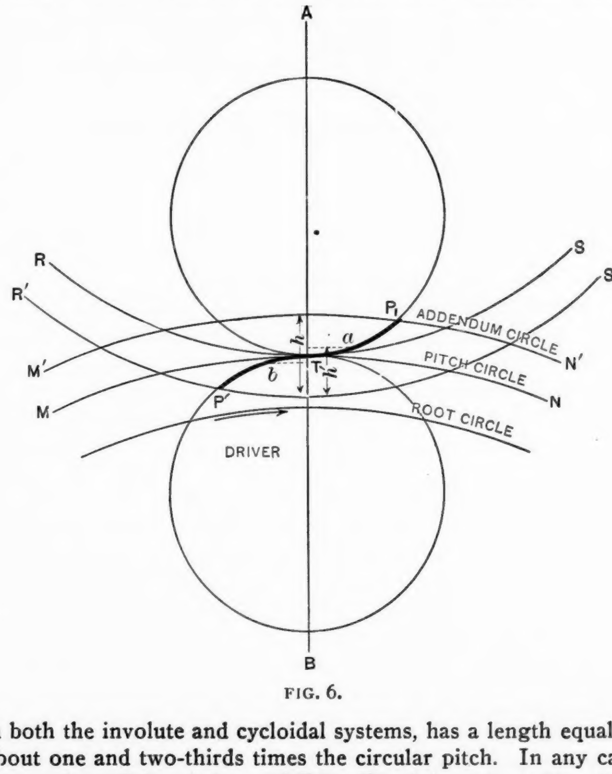


FIG. 6.

in both the involute and cycloidal systems, has a length equal to about one and two-thirds times the circular pitch. In any case, by stepping off the pitch on P^1 T P_1, Fig. 6, from both P^1 and P_1, we shall obtain the points a and b, which indicate that with perfect gearing the two pairs of teeth will be simultaneously in contact at P^1 and at a, and will remain in contact until b and P are reached; the distance from b to a will be traversed with only one pair of teeth in contact; however, in this latter case. it will be noticed that the leverage of the tooth is very much reduced. For a 12-toothed pinion (cycloidal system) this leverage, shown in figure as h^1, is $.41\,p^1$; and for a rack it is $.35\,p^1$. Under these conditions, then, the dimensions of the tooth would be determined upon the supposition that the whole working load is carried by one tooth, and that the force is applied with a leverage of $.35\,p^1$ to $.41\,p^1$, the constant depending upon the number of teeth. For the 15° involute system the effective height for a 12-toothed pinion is the same as for the cycloidal system, viz.: $.41\,p^1$; but for a 15° involute rack there are always two teeth in contact.

This determination is correct for all gear teeth properly shaped

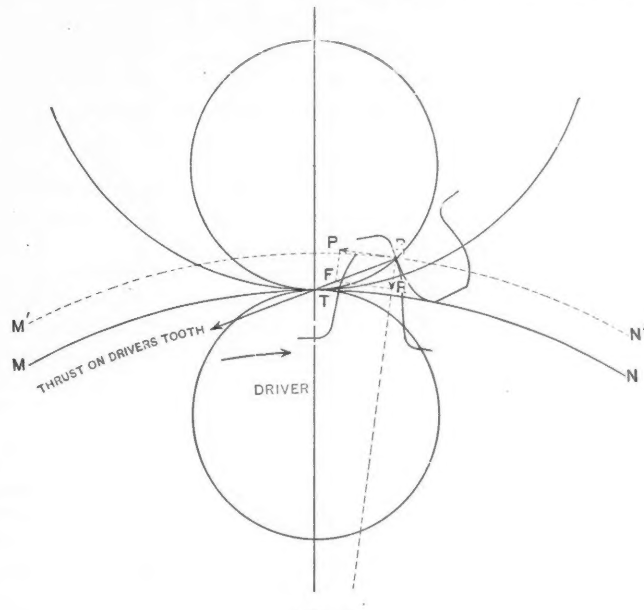


FIG. 7.

and spaced, but as Mr. Wilfred Lewis has so ably pointed out, it must be admitted that mechanical perfection in forming and spacing has not yet been reached, and that the slightest deviation in either respect is sufficient to concentrate the entire load at the end of one tooth. Even with cut gears the same conditions obtain, for the cutters ordinarily employed are correct only for a single gear, although they are used within certain equidistant limits for various other gears. To what extent the elasticity of bronze, copper and steel influences the distribution of pressure on cut gears is not known, and it is, therefore, unsafe to consider its effect. As the teeth become worn the concentration of the load may be reduced, but until this wear takes place the whole load should be assumed as acting at the extreme end of the tooth.

For rough cast iron and mortise gears, as we have shown, it is possible for the whole load to be concentrated on the corner of one tooth, and this may occur even with the best cut gears, owing to careless alignment or lack of stiffness in the shafting and supports; but when iron patterns are used with accurately cut teeth, especially if the mold is baked, or when the gear is machine molded, or the teeth are cut, the contact should be fairly uniform and the load evenly distributed across the face of the tooth. It must be understood, however, that in many cases circumstances will arise when it is impossible to secure these favorable conditions; particularly is this true when the gearing is subjected to shock and variable loads. However, with careful work, stiff shafts and not too wide a face, the assumption of fairly uniform distribution of pressure across the tooth may be considered as satisfactory for general practice.

The pressure which comes upon the tooth in the direction of the line of thrust (the common normal to the tooth surface) is greater than that which transmits motion, for if we resolve the force, F, along and perpendicular to the radius at the point of contact, P_1, Fig. 7, the radial component, R, does not tend to produce rotation: it will, however, exert a pressure on the bearings, and tends to crush the tooth; as the compressive force has little effect upon the strength of the tooth, especially with cast iron gears, its influence may be neglected.

The other component, or P, acts at right angles to the radius and produces rotation by its pressure on the tooth and thus may be considered as the effective working load.

Now considering the tooth as a rectangular cantilever with the load P uniformly distributed across the outer edge, as in

Fig. 1, the thickness, t, assumed equal to $0.49\ p^1 - .02''$—for well made cast gears, may be obtained by equating the bending moment to the moment of resistance of the tooth; that is,

$$P\,h = f \times \frac{1}{6}\, b\, t^2 \quad (8)$$

Assuming the effective height of tooth, $h, = 0.6\,p^1$ (circular pitch); breadth of face, $b = x\,p^1$; thickness of tooth, $t = 0.49\,p^1 - .02$, therefore, $t^2 = .24(p^1)^2 - .0004$, in which we may neglect the constant, as it will not appreciably affect the result.

Substituting these values in (8) we have:

$$P \times .6\,p^1 = \frac{f\,x\,p^1 \times (.24\,p^1)^2}{6}$$

$$P = f\,\frac{x\,(.24\,p^1)^2}{3.6} = 0.066\,f\,x\,(p^1)^2 \quad (9)$$

$$p' = \sqrt{\frac{P}{.066\,f\,x}} = \sqrt{\frac{15\,P}{f\,x}} \quad (10)$$

For cut gears the addendum is usually made equal to the reciprocal of the diametral pitch, or $1/p$; but since $p\,p^1 = \pi = 3.14$ we have $1/p = 0.32\,p^1$, hence the effective height of tooth now becomes $0.64\,p^1$. The thickness of tooth in this case is $0.5\,p^1$; therefore the load, from (8) is:

$$P = .065\ f\,x\,(p^1)^2; \text{ and}$$

$$p' = \sqrt{\frac{15.4\,P}{f\,x}}$$

results practically the same as those just obtained. If the breadth of face equals twice the pitch,

$$\text{or } x = 2, \quad \text{then } p' = 2.74\sqrt{\frac{P}{f}};$$

$$\text{if } x = 2\tfrac{1}{2}, \quad \text{`` } p' = 2.45\sqrt{\frac{P}{f}};$$

$$\text{`` } x = 3, \quad \text{`` } p' = 2.24\sqrt{\frac{P}{f}};$$

$$\text{`` } x = 3\tfrac{1}{2}, \quad \text{`` } p' = 2.07\sqrt{\frac{P}{f}};$$

It will be noticed that the formulas thus deduced apply only to rectangular teeth, as in Fig. 1, whereas in practice we have to deal with such forms as given in Figs. 3 and 4. In this case the width of tooth is considerably less measured on the working depth circle than on the pitch circle, and fracture would occur between the points a a, where the thickness is t^1.

With a uniformly distributed pressure, P, acting at the extremity of the tooth as shown, the relative strength of this tooth compared with one of rectangular section having the uniform thickness t varies as $\left(\frac{t'}{t}\right)^2$. Thus if a gear tooth of 1 diametral pitch ($p^1 = 3.14$) measures 1.52″ on the pitch circle, and 1.34″ on the working depth circle between the points a and a, Fig. 3, the strength of the tooth will be only $\left(\frac{1.34}{1.52}\right)^2 = 0.78$ of that which it would have if the tooth had the constant thickness, 1.52″. In the same way, if the thickness of tooth at a a, Fig. 4, measures 1.76″, its relative strength will be $\left(\frac{1.76}{1.52}\right)^2 = 1.34$.

This suggests a convenient method not only of ascertaining the load which a given gear will sustain, but also one which will give a suitable pitch when the load and number of teeth are known.

(To be continued.)

* * *

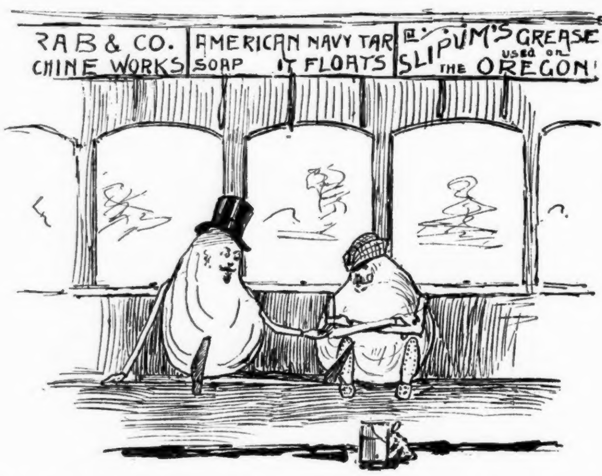
"Locomotive Engineering" describes a rack for the tool room that is in use at the Wilmington shops of the Philadelphia, Wilmington & Baltimore Railroad, and which is capable of unlimited expansion. It is described briefly as consisting of a series of columns made of ½-inch tubes extending from floor to ceiling in the center of the tool room, so as to be accessible from all sides. The tubes are spaced about 15 inches apart each way, and are connected together by flat iron ¼ x ¾ inch, at intervals of, say, 6 inches, by U-bolts encircling the tubes and passing through the flat. On these flat bars rest cast-iron pans or trays, about ¾ inch deep, flanged on all sides, and wide enough to just fill in between the tubes.

THE SHELLFISH CLUB.

A TRUE FABLE, BY THE LOBSTER.

'Twas Saturday night, and the Clam was returning from the beach with his marketing squeezed under a seat in the hot end of a trolley car, thankful that another week's wages had been earned to pay his rent and feed the many little Clams that gamboled around his fireside; also envying the other Shellfish whose lots in life seemed to be more pleasantly cast, when who should bustle up and squeeze in beside him but the Oyster. The Oyster and the Clam had been shopmates together years ago in old Sand-Crab's "devil-mill," making head-stocks, tail-gates and fifth wheels. They often sat on the work bench during noon hours building castles in the sand of future prosperity and riches, while eating their cold salad lunches; but the Oyster, with a pluck common to his race, had left the work bench and embarked in business for himself, and was now running a devil-mill of his own. He had grown wealthy, owned a fine castle upon the rocky part of the shore, and did little else but live on jelly fish and whale oil and travel in Pullman cars.

As they chatted over old times and recalled incidents of their early lives, the Clam asked the question: "Oyster, old boy, it must be mighty nice to be rich like you and have everything you want, besides all the things you don't want, thrown in, for yourself and family? How you must enjoy it?" The Oyster paused thoughtfully before answering, and then said: "I don't know about that; there are times and times when I look back with a heartache, my boy, to the days when you and I wore greasy overalls and filed cuttle bones and cut shark teeth for old Sand-



Crab. Then, you remember, I used to pinch a little out of my week's pay every Saturday night, and keep all the extras made during the week from overtime and put it by, so when the wife's birthday came around I could take home a nice, bright-flowered linsey-woolsey gown from Wanamaker's bargain counter of sea-fan goods, and after our evening meal the madame and my six little Oysts would gather around the fireside and admire it. Of course, it was only a cheap fabric, but it marked one of the bright spots in our family annals, and was a topic of conversation for days; and when, on the following Sunday Mrs. Oyster and I took our afternoon promenade on the beach, we were the cynosure of every eyeless fish on the block. At other times, on a holiday I would manage to hire Sand Fiddler's team, and we would all take a long spin up to Rockaway, and that would be another incident to be chatted about during the evenings, and many were the hugs and kisses that Daddy Oyster received. But now it is different. The Madame has so many fine dresses that I wouldn't know what to buy, and when I go home of an evening the little Oysts, grown to manhood, are out on their new bicycles, and the Madame is out with the carriage calling on Mrs. Clam Chowder or some one else, and your humble servant eats his cold lunch of salad or whatever else the servant sees fit to give him, and after wandering about the house awhile betakes himself off to the Poker Club for an evening's entertainment. To tell the truth, my dear Clam, wealth, like poverty, has its misfortunes, and it is hard to tell which of us to-night has the best lot in life." So saying, he proferred the Clam a cigaroot and left the car.

Did you ask what became of the Clam? We don't care what became of the Clam, it was the Oyster's story we wanted. The moral? There isn't any.

THE CENTER OF GRAVITY.—2.

PLANE FIGURES CONTINUED, AND THE CENTER OF GRAVITY OF SOLID BODIES.

BENJ. F. LA RUE.

IRREGULAR FIGURE; RECTANGULAR CO-ORDINATES.—The center of gravity of any irregular figure bounded by straight lines may be found by dividing it into triangles, finding the center of gravity of each triangle and then finding the center of gravity of the system of triangles, the area of each being considered to be concentrated at its center of gravity. For finding the center of

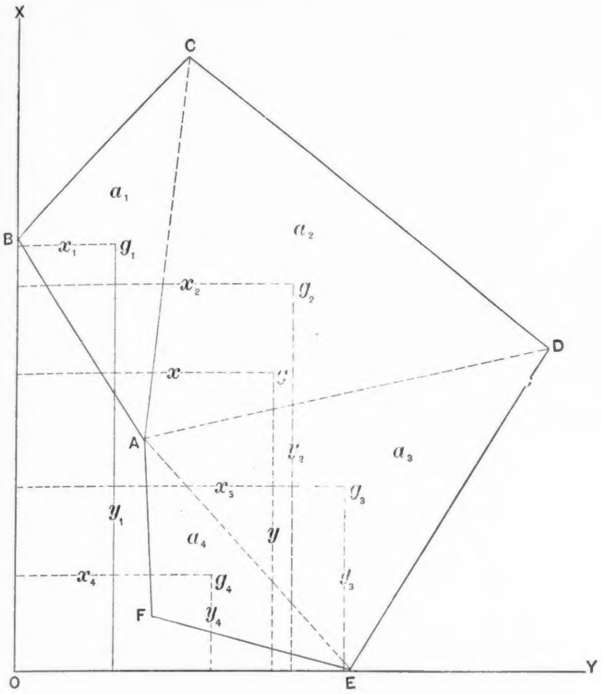


FIG. 9.

gravity of the system of triangles, the method of rectangular co-ordinates may be employed. Let *ABCDEF*, Fig. 9, be any irregular figure bounded by straight lines. By the lines *AC*, *AD* and *AE*, the figure can be divided into the four triangles *ABC*, *ACD*, *ADE* and *AEF*, whose centers of gravity g_1, g_2, g_3 and g_4 may be found by the method explained for triangles. Draw the vertical and horizontal axes *OX* and *OY*, intersecting at *O*; these may be any vertical and horizontal lines, but it is generally convenient to draw them through the left-hand and lower extremities of the figure, as shown; *OX* is the axis of abscissas and *OY* is the axis of ordinates. The lines x_1, x_2, x_3 and x_4 are, respectively, the abscissas of the centers of gravity of the several triangles or their perpendicular distances from the axis of ordinates; and the lines y_1, y_2 y_3 and y_4 are, respectively, the ordinates of the same points, or their perpendicular distances from the axis of abscissas. If a_1, a_2, a_3 and a_4 represent the areas of the four respective triangles, then the abcissa x to the center of gravity g of the entire figure will be given by the equation:

$$x = \frac{a_1x_1 + a_2x_2 + a_3x_3 + a_4x_4}{a_1 + a_2 + a_3 + a_4};$$

and the ordinate y to the same point will be given by the equation:

$$y = \frac{a_1y_1 + a_2y_2 + a_3y_3 + a_4y_4}{a_1 + a_2 + a_3 + a_4}.$$

This method applies to any figure or system of figures, either separate or joined, that can be divided into triangles or other simple figures such that their centers of gravity and areas can be determined. It will also apply to a system of weights or solid bodies.

IRREGULAR FIGURE; EQUILIBRIUM POLYGON.—The gravity axes and, consequently, the center of gravity, of a very irregular figure can be determined by the equilibrium polygon, as constructed for determining the position of the resultant of any system of forces. Let *MN*, Fig. 10, be any plane figure whose center of gravity is to be determined; the following construction may be employed:

By means of parallel lines, divide the figure into a sufficient number of parts of equal width, so that each part will be approximately a trapezoid, as shown by full lines in the figure; also draw lines midway between these dividing lines and parallel to them, as shown in the figure by dotted lines; as the trapezoids are of equal width, these dotted lines will be proportional to their areas. On a vertical line, as the line 0—6 (commonly called the *load line*), lay off in order downwards the amounts 0—1, 1—2, etc., by any scale, equal, respectively, to these latter lines, that is, proportional to the areas of the respective trapezoids. Choose any point *P* as a pole and draw the radial lines *P*—0, and *P*—1, etc. Find the centers of gravity *a*, *b*, *c*, etc., of the respective trapezoids, and from these centers draw lines vertically downward, as the lines *aa'*, *bb'*, etc. Through any point on the line drawn downward from *a*, as the point *a'*, draw the line *a'g'*, of indefinite length and parallel to *P*—0; from the same point also draw the line *a'b'*, parallel to *P*—1, to intersect the vertical through *b*; from this intersection draw the line *b'c'*, parallel to *P*—2, to intersect the vertical through *c*, etc. The line *e'f'*, drawn parallel to *P*—5, intersects the last vertical at *f'*; from this intersection, draw the line *f'g'*, parallel to *P*—6, to intersect *a' g'*; the vertical line *gg'* drawn through the intersection *g'*, will be a gravity axis of the figure. In a similar manner the figure may be divided into another set of trapezoids of uniform width, by parallel lines extending in a different direction, and the position of another gravity axis determined; the intersection of the two gravity axes will be the center of gravity of the figure.

The method of construction just described is generally the most expeditious, but it is not necessary that the lines dividing the figure be drawn at equal distances apart, nor that they be parallel; they may divide the figure into smaller figures of any form, of which the centers of gravity and areas can be easily determined. It is necessary only that each of the vertical lines *aa'*, *bb'*, etc., be drawn through the center of gravity of the corresponding part, and that the amounts 0—1, 1—2, etc., laid off on the vertical load line, be proportional to the areas of the respective parts.

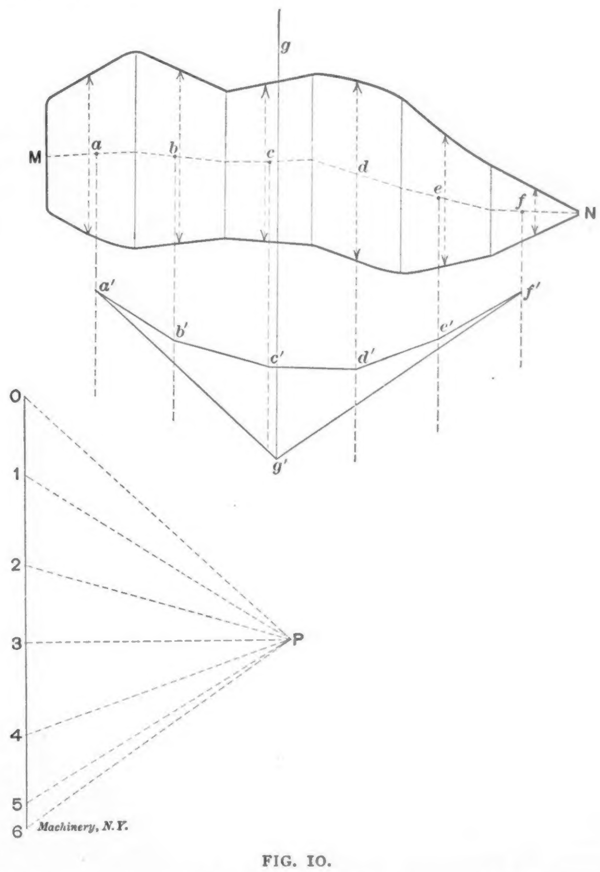


FIG. 10.

ANY PLANE FIGURE; EXPERIMENTAL METHODS.—To find the center of gravity of any plane figure experimentally, proceed as follows:

First Method.—Cut the figure out of cardboard or very thin metal plate. Balance it across a perfectly straight knife edge and mark the line of the knife edge across the figure; then turn the figure about 90 degrees and balance it across the knife

edge again, marking the line of the knife edge. Each line defining the position of the knife edge will be a gravity axis, and the intersection of the two axes will be the center of gravity of the figure.

Second Method.—Cut the figure out of cardboard or thin metal plate, as in the preceding method. Suspend it by a point near one edge in such manner that the figure will be free to turn upon the point of suspension; suspend a plumb line from the same point of suspension and mark its direction across the figure. Then suspend the figure from some other point and again mark the direction of the plumb line. The two lines defining the positions of the plumb line will be gravity axes, and their intersection will be the center of gravity of the figure. The figure may be suspended by a pin driven through the cardboard into a vertical wall; but, in order that the figure may be free to turn, it will generally be better to suspend it by a thread.

Center of Gravity of Solid Bodies.

The center of gravity of a sphere, spheroid, cylinder, cylindrical ring, cube, prism, parallelopipedon or any regular polyhedron is at the geometrical center of each body.

The center of gravity of a cylinder or prism is at the middle point of a line joining the centers of gravity of its two parallel surfaces.

The center of gravity of a hemisphere is in its axis, or radius perpendicular to its base, at ⅜ the length of the radius from the center of the sphere.

The center of gravity of a right cone or right pyramid is in the line joining the vertex with the center of gravity of the base, at ¼ the length of the line from the base.

The center of gravity of a body is not always within the body itself.

If a body be suspended freely at a point other than its center of gravity, its center of gravity will be vertically below the point of suspension. This principle affords an easy method of finding the center of gravity of any body, as described in the second method for finding experimentally the center of gravity of any plane figure.

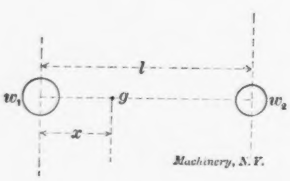


FIG. 11.

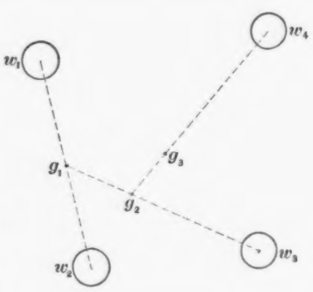


FIG. 12.

System of Two Bodies.—The center of gravity of a system composed of two bodies is in a straight line joining their respective centers of gravity, and at a distance x from the center of gravity of the body, of the weight w_1, the value of x being given by the following equation:

$$x = \frac{w_2 l}{w_1 + w_2}$$

in which w_1 and w_2 are the weights of the two respective bodies, and l is the distance between their centers of gravity. See Fig 11.

System of Two Bodies Connected by a Material Bar.—If the centers of gravity of two bodies are joined by a material bar of uniform cross-section, the center of gravity of the system will be in the bar at a distance y from the center of gravity of the body of the weight w_1, the value y being given by the equation:

$$y = \frac{l\,(w_2 + \tfrac{1}{2}b)}{w_1 + w_2 + b}$$

in which w_1 w_2 and l represent the same values as in the preceding equation, and b is the weight of the bar.

System of Three or More Bodies.—The center of gravity of a system of three or more bodies may be determined by first finding the center of gravity of any two of the bodies, then, considering the combined weight of these two bodies to be concentrated at their center of gravity, finding the center of gravity of these two and a third body; and continuing the process until the center of gravity of the entire system is found. Thus, in Fig. 12, g_1 is the center of gravity of the two bodies w_1 and w_2, found as for any system of two bodies. Now, considering the combined weight of w_1 and w_2 to be concentrated at g_1, the center of gravity of g_1 and w_3 is found in the same manner; the center of gravity of the system of three bodies w_1, w_2 w_3, being thus found to be at g_2. Next, considering the combined weight of w_1, w_2 and w_3 to be concentrated at g_2, the center of gravity of g_2 and w_4 is found, giving g_3 as the center of gravity of the system of four bodies w_1, w_2, w_3 and w_4. In the same manner the method can be extended and applied to any number of bodies. It can also be applied to any irregular plane figure, by dividing the figure into triangles, as previously explained for the method of rectangular co-ordinates. It will be readily understood that the latter method can also be applied to any system of solid bodies, by simply substituting the weights of the bodies for the areas of the triangles.

* * *

A GROUP OF OLD TIMERS.

Editor Machinery: I send you a few photographs that seem to me to illustrate an oft-made assertion, "The human animal loves ornament." (See opposite page.)

These photographs were made during the summer of 1897, and all the machines shown were in actual use at that time. It seems to me needless to suppose that all the ornamentation shown on some of these machines was put there "just for fun." That it helped to make sales, is, I believe the true reason for it all, and I think we can see the same tendency to-day in the embossed picture work on sewing machines, bicycles, typewriters, etc.

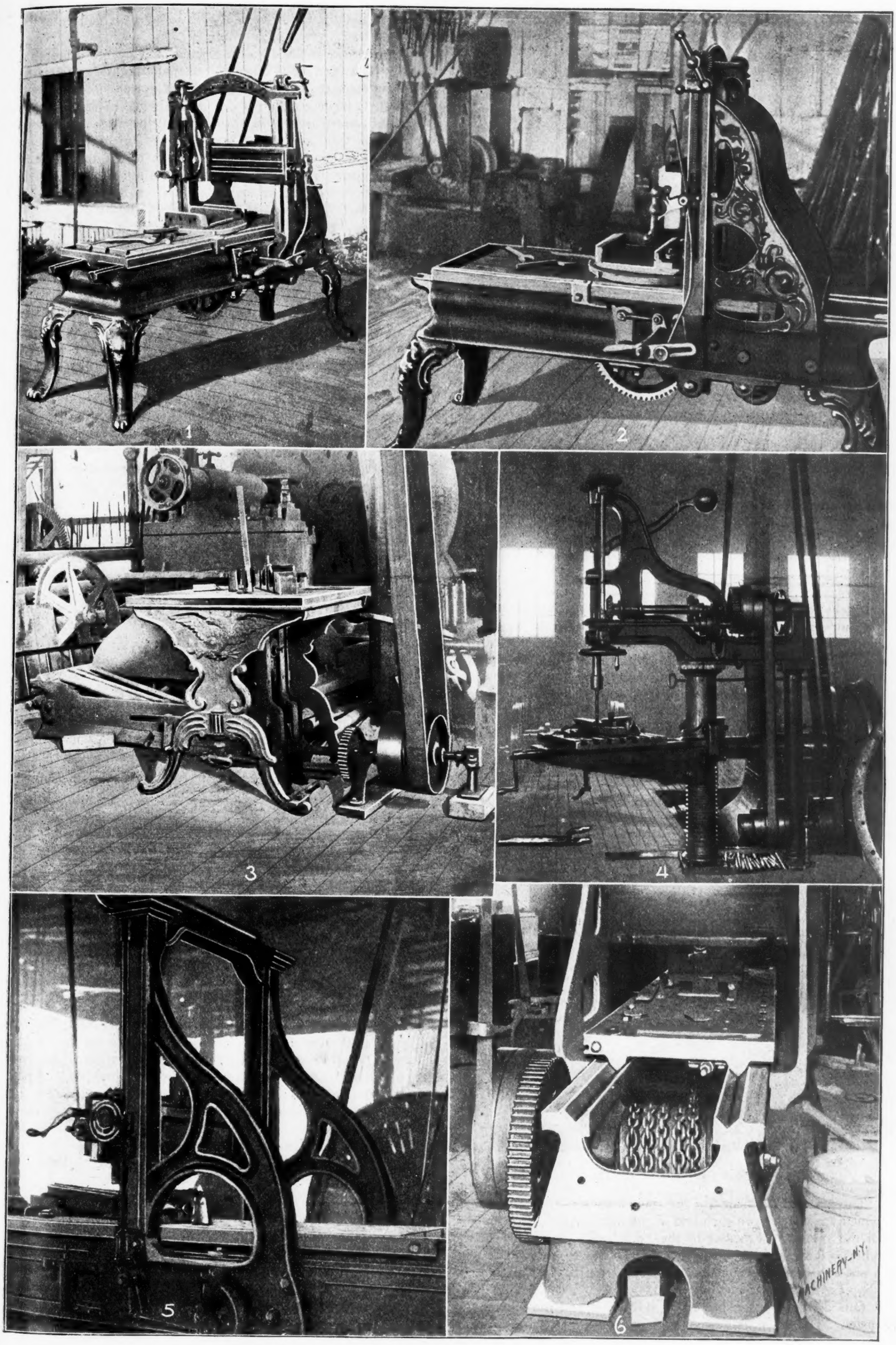
The grease and dirt of the shop made it difficult to keep such work clean, and good sense has now shown it to be needless and costly on machine tools. Fig. 1 shows what is sometimes called the piano style of planing machine. The legs and bed show a marked resemblance to the old-fashioned square piano. This is the most ornamented of any planing machine I have yet seen. Fig. 2 shows the uprights of the same machine, and as the machine was very old, all the scroll work must have been hand carving and not clay work. This planing machine was built by L. E. Osborn & Co., New Haven, Conn. It has changed hands so many times I was unable to learn its age, but it is a "before the war" relic. It is in use in a Cleveland, O., shop. Fig. 3 is a view of a key-seating machine in use at Painesville, O., and is of the same general style of architecture as the planing machine shown in Figs. 1 and 2. I could learn nothing of the age of this machine, or who its makers were. I have no doubt of its being as old as the above planing machine, and it shows many signs of being built by the same firm.

Fig. 4 shows a change in design, and is a drill press in use at Cleveland, O. Fig. 5 shows a planing machine in use at the same shop as Fig. 4, and they have indications of being from the same designer. They show an attempt to brace a surface, that is, at right angles to the main bed of the machine, trying at the same time to make it ornamental by the use of curves. This design is somewhat after the style of the ordinary cast shelf bracket. Fig. 6 is a view of a planing machine no doubt common enough fifty years ago, but is the only one left, so far as I know, having a chain as a driving mechanism. This machine was evidently designed for utility, and that alone. While the main bed is quite rigid, the uprights are weak through faulty design. It was in use in the old "Cuyahoga Works," now the Cleveland Shipbuilding Co., and is the only old machine in use among a fine lot of modern tools.

Howard P. Fairfield.

* * *

An engineer writes us, stating among other things, that his employer not only will not buy him an indicator, but will not let him drill and tap the cylinder, so that he can put on an indicator that he is able to borrow. The reason given is that he knows the engine is throwing away coal, and does not care to know more about it. As this man evidently has money to burn, and the engineer has a good-sized family to support, we would suggest that it would be a charitable deed to find out how much surplus change is going into the fire every day, and give the engineer a chance to reclaim a liberal percentage of it. In other words, buy him an indicator and adopt the premium plan.



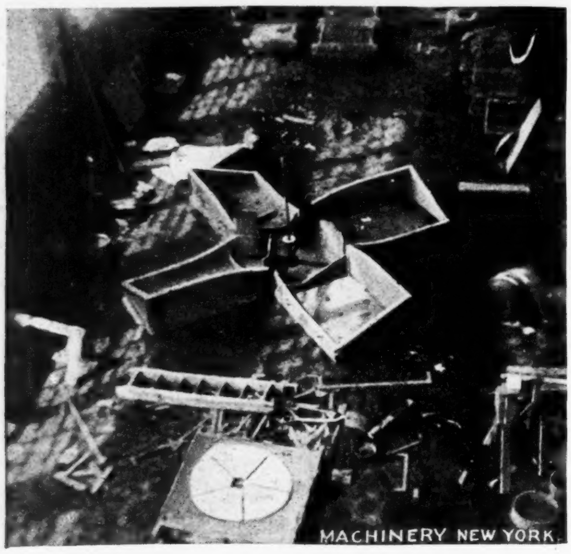
A GROUP OF OLD TIMERS. (See opposite page.)

MOLDING A PROPELLER WHEEL.

DRY OR GREEN SAND.

Geo. W. Dean.

In molding a propeller wheel the pattern of one blade and a quarter of the hub is generally used. A spindle, from 2 to 3 feet in length, the diameter measuring not less than 1½ inches, is tapered at one end, in order that it may fit into a socket or base. A simple way to make a base is to take a coupling pattern and mold it up in open sand.

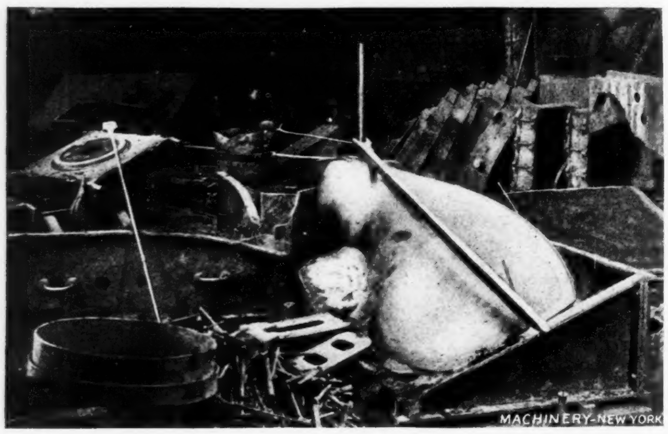


NOWELS IN POSITION.

A heavy coat of lead is applied to the tapering end of the spindle, and thoroughly dried. It is placed upright in the mold, and iron is poured around it. After the iron is set the spindle can be removed, leaving a base suitable for the purpose. The spindle and base are set up in position, the base being placed below the surface of the floor. An arm is fastened to the spindle, to which a straightedge is attached.

A bed is swept up on the floor about 2 feet larger than the diameter of the wheel, or large enough to allow the flask to get an even bearing. The bed is divided off into as many sections as there are blades to be in the wheel.

A center-mark is made on the outer edge of each nowel, and in placing the nowels in position, these marks must come directly over the section marks made on the bed. The nowels are rammed up within 2 or 3 inches of the pattern.



SHOWING STRAIGHT EDGE AND PATTERN IN POSITION.

The pattern is attached to the spindle by means of two plates, which are fastened on each end of the hub, having holes the size of the spindle, the center of the holes being the center of the wheel. It is necessary to have a collar fitted to the spindle with a set screw and placed beneath the pattern to prevent it from slipping down on the spindle.

One of the copes is placed on the nowel and the pattern adjusted. The collar is pushed up against the pattern and firmly screwed, after which the cope is removed. If the spindle is not fastened at the top, it will be necessary to level the pattern at the hub.

A straightedge is fitted to the spindle, having one edge on a line with its center. The other end of the straightedge is placed to the center-mark that was made on the nowel.

A mark is made on the pattern in line with the straightedge. The pattern must be kept in position until the nowel is rammed up.

The cope is next put on and rammed up in the ordinary way. The sand is struck up level with the top of the hub in order that a covering core may be placed on it when the mold is put together. The cope is then removed, the necessary venting done and the pattern is drawn from the mold.

If the mold is to be dried, stakes are driven around the nowel for guides. Afterwards this part of the mold can be removed to the oven. The pattern is placed over the spindle and dropped on to the collar and swung around to the next nowel. The straightedge is placed against the spindle and the other end placed across the mark on the nowel. The pattern is put in position by bringing the mark on it in line with the straightedge. The pattern is leveled at the hub and is now ready to ram up as the blade already mentioned.

The same process is repeated in all additional blades.

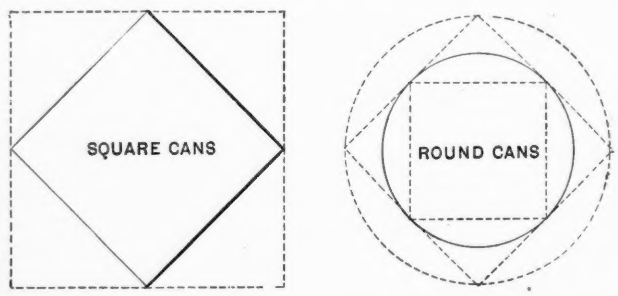
After each mold has been thoroughly dried, it is returned to its former position. The spaces between the blades at the hub are filled in. The cope and nowel are fastened together by means of bolts, the use of clamps not being considered as safe. The spindle is now removed, and a piece of waste is placed in the socket.

The lower print is formed for the center core, which is next put in. The covering core having one or two holes for runners and riser, also print for center core, are now placed over the top of the hub. These cores are held in position by weights. The sprues being adjusted, the runner is made up in the ordinary way. The mold is now ready to be poured.

* * *

SIZING UP TIN CAN DIES.

Here is a sketch that has become familiar to many as the solution of a puzzle, and it is gratifying to find that it has been put to a more practical use—shop use. A Canadian mechanic called the writer's attention to it last winter, in connection with die work. In making dies for tin can work he often had to make one size of the same depth, but of just twice the holding capacity of the other, and for square cans the size of either can be found



from the size of the other by drawing the two squares full size and in the relation to each other shown in the sketch,—corners of one at the middle of the sides of the other. In the case of round cans, if a square is inscribed within the circle and enlarged in the same way, a circle drawn through the corners of the larger square will represent the size of the larger can. Of course the same thing can be done with a few figures, and some would prefer to do it that way, or else consult some table; but somehow such things don't appeal to certain men and the graphical method is easily remembered and always at hand.

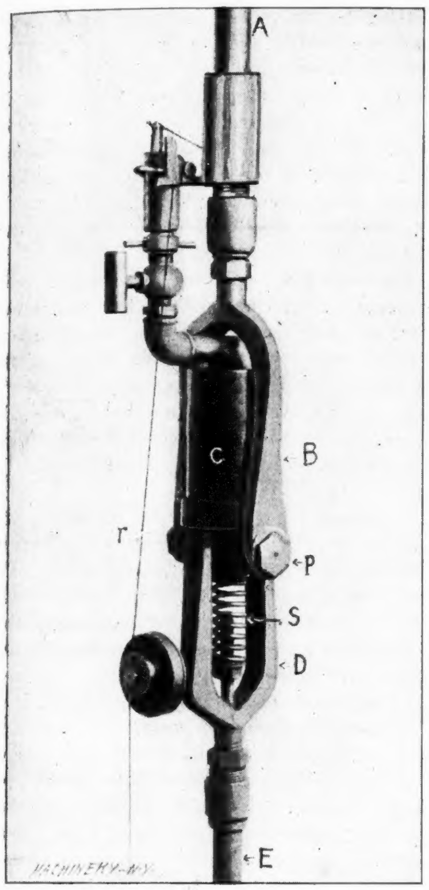
* * *

Not long ago we were present at a shop where a machine tool was being unpacked, built by one of the leading firms of this country, whose reputation for good work has been built up solely by the excellent quality of the machines turned out. In taking the machine apart for cleaning, one of the first things that showed up was a prick-punch fit. It was not a half-hearted affair, but a good, generous one, all the way around. This did not diminish our respect for the products of this firm, however, for accidents will happen in the best of regulated concerns.

A RECIPROCATING DYNAMOMETER.

O. P. Hood.

While on the subject of dynamometers the readers of Machinery may be interested in a form of the hydraulic dynamometer, adapting it to such work as determining the stresses in reciprocating pump rods. The problem proposed was to measure the stresses in reciprocating rod of pumps used for raising water by means of windmills. Here the velocities are not high, but the stresses are great and varied. The only reciprocating dynamometer available on the market was one prepared by Mr. Giddings, which employed flat springs and gave an autographic card, in which compression stresses were shown in the same manner as tension stresses. This was an objection and also great difficulty was found in obtaining springs of sufficient range to satisfy the needs. A Flather dynamometer was used to drive the whole mechanism, but because of the belt speed necessary to do the work, the cards drawn were so small for each stroke of the pump that peculiar characteristics of the card were difficult to detect and interpret. The card included also the losses due to driving mechanism, which were not wanted. The cut will show the device built for the purpose, and which gave very satisfactory diagrams. A is the upper part of the pump rod extending into a fork B. On a pin (p) extending across the tines of fork B, a plunger was carried extending well up into the cylinder C. Cylinder C was screwed into a ring (r) forming part of the tines of an inverted fork D, which was an extension of the lower section of the pump rod E.



It is evident that oil being interposed between cylinder and plunger, stresses of tension in the lower part of the pump rod would be transmitted to the upper part of the rod through compressed oil, and that the varying pressure of oil could be measured by an indicator the same as in a Flather dynamometer. A suitable reducing device gives a drum movement proportional to the pump stroke and the card drawn is entirely similar to a card from a steam engine cylinder. Such a card gives similar information as to the sequence of events in the pump cylinder and gives information as to the action of valves difficult to otherwise obtain. Computations as to actual HP. used by the cylinder are also easily made.

The plunger was made of cast iron, and was carefully ground to fit a cast iron cylinder 10 square inches in area. The plunger was about 5 inches long and without packing of any kind. This stood pressures of 200 pounds per square inch without serious leakage. Leakage was easily replaced by a small hand pump connecting with the fittings carrying the indicator. These high pressures are only momentary in pump work, and the device succeeded better than would probably be the case where pressures are long continued in one direction. Water was found too limpid, cylinder oil too viscous for successful work, and a light machine oil was used. As in the Flather dynamometer accuracy is greater where the variations in pressure are accompanied by a slight jar. There is no lack of sufficient jar in pump work. To enable the indicator to record compression in the rod, as well

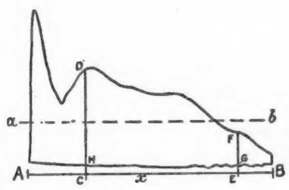


FIG. 1.

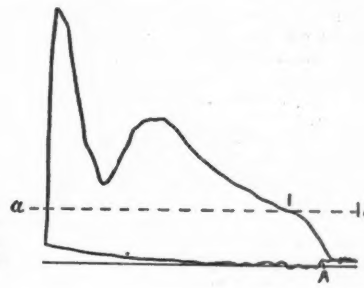


FIG. 2.

as tension an initial compression was put upon the oil by means of the adjustable spring S. This spring forced the plunger into its containing cylinder and raised the "zero" line drawn by the indicator pencil. As the maximum movement of the plunger was less than 4-100 inch, this initial compression could be considered constant for the spring (S). A tension in rods A and B increased the initial compression of oil, while compression between A and B simply lessened the initial compression on the oil.

Fig. 1 is a typical card from a simple pump.

Distances above A and B represent tension in the rod in pounds, while distances along A B represent positions in the length of stroke. The line a b represents the static load of the water, the diagram falling below this line at the end of the stroke. Fig. 2 shows a diagram from a pump having such a speed that water continues rising through both valves until the piston is well on its downward stroke at A.

Fig. 3, a diagram from a high-speed pump, shows the time of closure of the valves at each end, and also compression in the rod at the beginning of the down stroke.

Fig. 4 shows still tardier closing of valves. Fig. 5 is from a pump having a high suction lift, the piston leaving the water in the cylinder at the beginning of the up-stroke and meeting the slowly rising water on the down stroke at the point X.

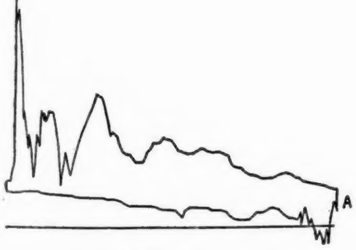


FIG. 3.

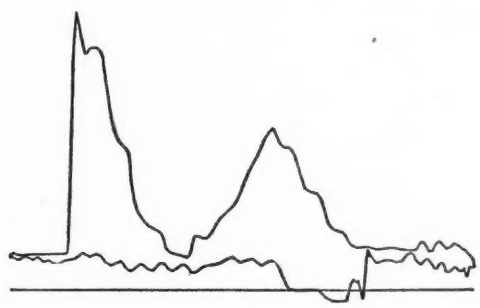


FIG. 4.

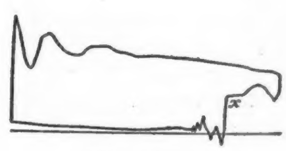


FIG. 5.

In 1873 Krupp's works exhibited at Vienna a crucible steel casting weighing 52½ tons. It was poured from 1,800 crucibles, each weighing about 60 pounds. Since this year many larger castings have been made.

* * *

The Engineers' Club, of New York, made a fitting recognition of the brilliant record of the engineering staff of our victorious fleet by tendering a dinner to the engineer officers during their stay at this port.

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OCTOBER, 1898.

Some of the engineering papers have much to say upon "educated engineers" vs. "practical" men, to the detriment of the latter, while many manufacturers express a very decided preference for the practical man. We believe in the educated engineer, but he must be educated in practice as well as in theory. We also believe in the practical man, only he must try to obtain enough education to apply his practical information intelligently. Either an educated man without practice or a practical man without education is bound to be more or less one-sided, and he should try to make up the dificiency as far as possible.

* * *

ROTARY ENGINES.

Several times questions have been submitted to us by men who were engaged upon the rotary engine problem. It is a problem that many mechanics become interested in at one time or another, even if they do not reach the point of making a model or taking out a patent. Patent attorneys have found the rotary engine field a prolific one to till, and many seeds have been sown there by persons too poor to afford either the time or money involved, who have in nearly every case failed to reap any harvest at all.

Some of the work upon rotary engines has been by men entirely conversant with the requirements of a successful steam engine, and who knew the results to be expected from an engine of the rotary type. There can be no criticism of them or their work, because they have attacked the problem with open eyes. But where there has been one man of this stamp, there have been hundreds who have had no knowledge whatever of the first principles of steam or the steam engine, and it is probable that the majority of these men have started with the idea that a steam engine of the usual type is wrong in principle, and that a large saving in power would be effected by substituting a simple rotary motion for the reciprocating motion of the piston, cross-head and connecting-rod of the ordinary steam engine. They may have heard also that the best steam engines waste most of the heat energy in the steam, and so have imagined that there was an opportunity for somebody to make a grand discovery that would revolutionize steam engine practice and win fame for the discoverer. To help dissipate notions like these, it will do no harm to point out a few facts like the following: With any type of engine there is a necessary loss in power due to the friction of the moving parts, and in the rotary engine this might and it might not be less than in the reciprocating engine, according to the construction. We have seen rotary engines designed that would have a much greater friction loss than the other type. Aside from this matter of friction, there is no loss of any account between the steam in the cylinder and the flywheel on the shaft, regardless of the style of engine, reciprocating, rotary or otherwise. There is no loss in the crank motion, because it can be proved that the amount of power put into any machine or mechanical movement, no matter what kind, is equal to the power given up by it, frictional losses neglected. This is a fundamental law of mechanics.

In fact, the common arrangement of the mechanism of a steam engine not only represents no mechanical loss, but has certain positive advantages that no other mechanism can have. The very fact that the piston slows down at each end of the stroke enables the steam to enter the cylinder under boiler pressure, before it is time for the valve to close, thus making it possible to derive the full benefit of expansion. Clearance and other items that have everything to do with the economical use of steam can also be regulated as desired.

With regard to the question of steam saving, it can be said that theorists and practical men alike have given it their attention and study for fully 100 years, and have reduced the subject to so much of a certainty that it can be told without much doubt where a saving will be made in the future and about what that saving will be. It is generally agreed that with the present materials of construction it will come along the lines of present practice through slight improvement here and there, such as using higher pressure steam and superheating. Moreover, it may be a surprise to some to know that no steam engine, however perfect, can by any possibility utilize more than a small per cent. of the heat energy in steam. High-duty pumping engines of to-day run on from 12 to 15 pounds of steam per H.P. per hour, and a perfectly constructed engine would use 10 or 12 pounds under the same conditions. The bulk of the loss in the use of steam is inseparable from the use of steam, and is not due to the crudeness of existing engines. Most of it would occur in the best engine that it would theoretically be possible to make.

In view of these facts and others that might be pointed out if space permitted, there does not seem to be the incentive for trying to improve the steam engine by the introduction of the rotary type that some have supposed. We do not mean by this that the rotary may not have a place to fill, and that it may not be able to fill it well; but it will be because of other reasons than that its mechanism and economy of steam are superior to the other form of engine.

It is undoubtedly true that under certain conditions and in certain places a good rotary engine would be a very desirable machine to have. There are certain constructive features about it that are difficult to overcome, and while they may not be insurmountable, they increase with the size of machine and probably preclude the possibility of a strictly rotary engine of large size being made a success. Questions relating to steam economy, like that of clearance or expansion, are not easy of solution with the rotary type, and here again is chance for trouble. We believe that small, compact, high-speed rotary engines may be found useful in some places, and especially so since the introduction of compressed air, but beyond this we should not like to predict that they would find any favor.

* * *

One of the hardest things to accomplish in shop management is to keep the work moving from the time it leaves the pattern shop until it reaches the erecting floor. A good plan is to insist that when an operation has been completed on a lot of pieces they are to be sent at once to the machine where the next operation is to be done. The work will go along faster if the pieces are waiting for a machine than if a machine is waiting for the pieces, and there is less liklihood of any of them being forgotten.

STEAM ENGINE TESTS.

METHODS OF CONDUCTING AND FIGURING STEAM ENGINE TESTS.

A. H. Eldredge.

Before beginning an engine test, we must first decide what are the objects of the test, and then select the apparatus necessary to conduct the work in order to attain the end in view.

Ordinarily the objects of an engine test are as follows:

1st. The indicated horse power, I.HP.

2d. The delivered horse power, D.HP.

3d. The water rate per I.HP. per hour.

4th. The regulation.

Besides these there are numerous other objects that may be desired, as the mechanical and thermodynamic efficiencies; the heat lost by radiation; the effect on the economy of steam jacketing, or of compounding, etc. The first three are the objects of most importance commercially, while the fourth ranks with them where a uniform rate of speed is desired, as for electric lighting.

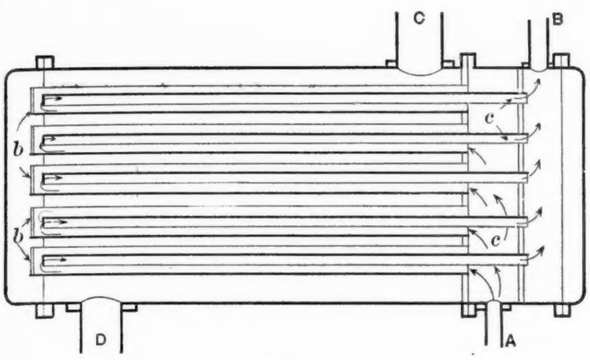


FIG. 1.

The apparatus necessary to determine the engine power and water rate consists of a steam engine indicator, an absorption brake, one or two calorimeters, one or more steam and common thermometers, two weighing tanks and platform scales for the brake and weighing tanks.

Measurements of Steam.

Two methods are commonly employed to measure the steam used by the engine: first by means of the surface condenser, and second by measuring the boiler feed. The surface condenser affords the most accurate way for measuring the steam used. It is a condenser in which the steam is kept separate from the condensing water, as shown in Fig. 1. Here the condensing water enters at (*A*), passes down through the large tubes (*b*), returns through the small tubes (*c*) and out at (*B*). The steam enters at (*C*), surrounds the outside of the large tubes, where it is condensed by their cooling surfaces. It then passes from (*D*) to the air pump, which discharges it into the weighing tanks to be measured. The temperature of the condensed steam should range between 90 degrees and 110 degrees F., and it will require approximately 25 pounds of condensing water per pound of steam.

The second method of measuring the steam is used when the engine is not provided with a surface condenser, and consists of measuring the water fed to the boiler. This is a more difficult task, but can be accurately done when proper care is taken. With this method we must use the boiler for no other purpose during the test. The feed to the boiler must be supplied through an injector, or if a steam pump be used the steam used by the pump must be supplied from some other source than the boiler supplying steam for the engine test, except where the exhaust from the pump can be turned directly into the boiler feed. The blow-off valve and all connections must be examined to see that they are perfectly tight, and the water level should be accurately measured before and after the test, and corrections made if necessary. It is important to run a test of this kind over a longer period of time than one where the surface condenser is used, and it should not be less than ten hours long.

Measurements of Power.

In all efficiency tests it is necessary to have some means of measuring both the I. HP. and the D.HP. The I.HP. is measured by the steam engine indicator, an instrument that gives accurately the pressure of the steam within the cylinder at all points of the stroke. In the present article we will not discuss the indicator, but simply how to connect it up, and how to figure the I.HP. from the diagram.

The method of attaching the indicator to the cylinder is shown in Fig. II. Care should be taken to drill the holes (*a*, *a'*) in the counterbore of the cylinder, where they cannot be closed by the travel of the piston. The connections should be as short and as direct as possible. By using a three-way cock one indicator is sufficient, taking a diagram first from one and then from the other end of the cylinder on the same card.

The next step is to reproduce accurately the motion of the piston at the indicator drum by means of a reducing motion of some form. One form, known as the pendulum reducing motion is shown at Fig. III. The pendulum swings from a pin (*A*) and receives its motion through pin (*B*) at the cross head. A string (*ee'*) is led to the indicator drum and should leave the pendulum at right angles to its central position.

Having determined the possible length of card from our indicator, we are able to locate the arm (*mm'*), Fig. III., knowing the stroke of the engine and length of the pendulum from the proportion,

$$\text{r} : \text{R} :: \text{Length of card} : \text{Stroke of engine}$$

$$\text{or r} = \frac{\text{Length of card}}{\text{Stroke of engine}} \times \text{R} \ldots\ldots\ldots\ldots (\text{I}).$$

Some means must be provided to stop and start the drum motion. One way is to hook back string (*ee'*).

There are two methods of working up the indicator cards, one by finding the area and the other by measuring the mean ordinate of the card. The area can be found with a planimeter; then knowing the scale of the indicator spring and the length of the card we have,

$$\text{Mean ordinate} = \frac{a}{l}; \text{ and M. E. P.} = \text{scale of spring} \times \frac{a}{l} . (\text{II.})$$

where (*a*) equals the area and (*l*) the length of the card.

If we do not have a planimeter, we can divide the length of the card into ten equal parts, then erect ordinates in the centers of these parts, and on a strip of paper lay off the consecutive lengths of these ordinates, divide this total length by ten for the

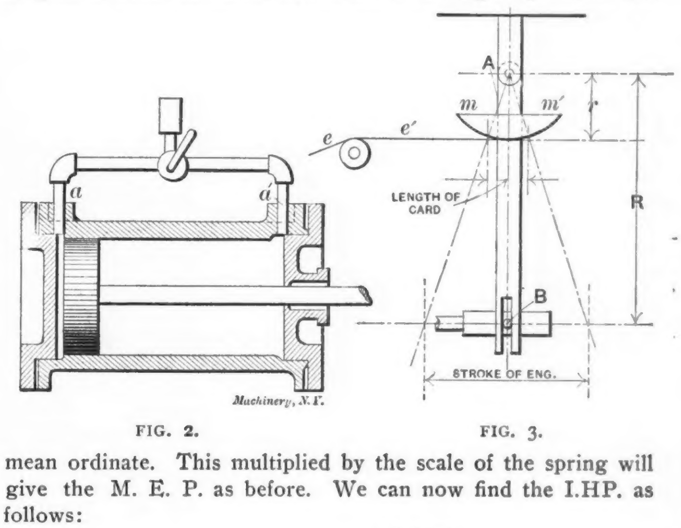


FIG. 2. FIG. 3.

mean ordinate. This multiplied by the scale of the spring will give the M. E. P. as before. We can now find the I.HP. as follows:

$$\text{I. H. P. head} = \frac{\text{p l A N}}{33\,000} \ldots\ldots\ldots\ldots (\text{III.})$$

$$\text{I. H. P. crank} = \frac{\text{p l (A} - \text{a) N}}{33\,000} \ldots\ldots\ldots (\text{IV.})$$

p = (M.E.P.) mean eff. pressure in lbs. per sq. in.

l = length of stroke in feet.

A = area of piston in inches.

a = area of piston rod in inches.

N = the number of revs. per minute.

The delivered horse power can be measured by an electrical resistance, but is commonly measured by some form of an absorption brake, as shown in Figs. (IV.) and (V.). The Prony brake Fig. (IV.) is a form commonly used. The body of the brake consists of two strips of band iron, to which are screwed hardwood rubbing pieces. An adjusting screw is provided at (*t*); also a brake arm securely bolted to one side, one end of

which rests upon a pair of platform scales. The rim of the wheel should be cast with flanges, and should be kept filled with running water during the test by some method, as illustrated in Fig. (IV.). Where the brake wheel is cast without flanges the writer has used wooden ones, which answered all purposes. Lubricate the face of the wheel with tallow or heavy oil. For the

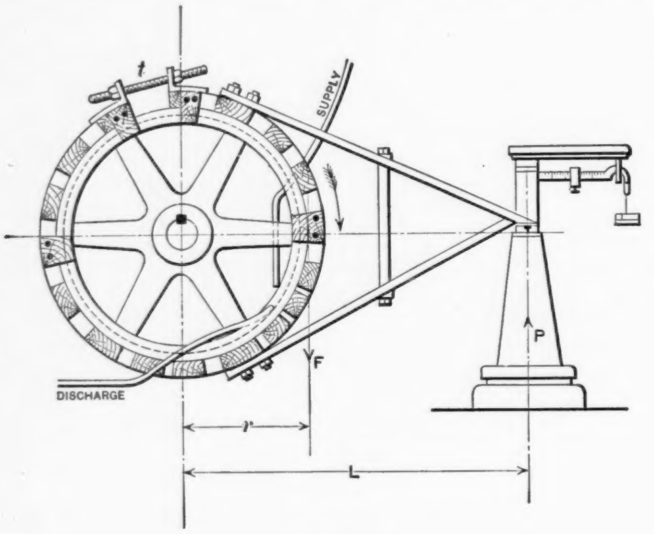


FIG. 4.

power absorbed by the brake in friction we will have as the moments about the center of the wheel Fr = PL (see Fig.), and the work in friction measured at the circumference of the brake wheel in foot-pounds equals

$$W = 2\pi r N F \text{ .... (V.)}$$

or at the scales

$$W = 2\pi L N P \text{ .... (VI.)}$$

and the delivered horse power D.HP. equals

$$\text{D. H. P.} = \frac{2\pi L N P}{33\,000} \text{ .... (VII.)}$$

$\pi = 3.1416$.
r = radius of brake wheel in feet.
L = length of brake arm in feet.
N = revs. of wheel per minute.
P = net scale reading in pounds.
One HP. = 33,000 pounds raised one foot in one minute.
$2\pi r$ = circumference of brake wheel.
$2\pi L$ = the distance in feet through which the load P would be carried were it free to move.

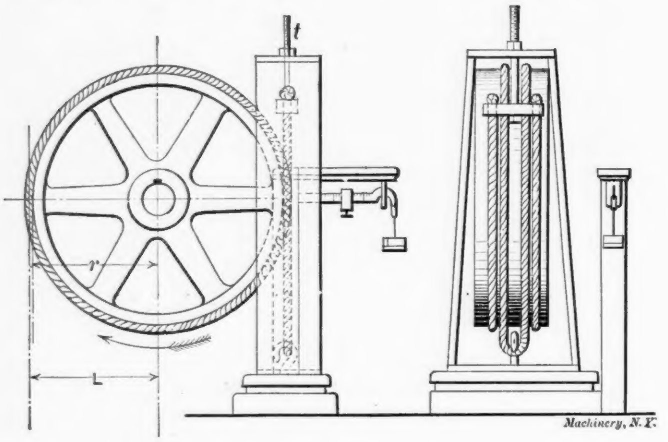


FIG. 5.

In figuring up the results the work can be easily shortened, as follows:

In equation (III.) we have $\frac{1\,A}{33\,000} = C$, a constant quantity.

In equation (IV.) $\frac{l\,(A-a)}{33\,000} = C'$, a constant, and in equation (VII.) $\frac{2\pi L}{33\,000} = C''$, also a constant, so we are able to write

$$\text{I. H. P. head} = C\,N\,p \text{ .... (VII.)}$$
$$\text{I. H. P crank} = C'\,N\,p \text{ .... (IX.)}$$
$$\text{D. H. P.} = C''\,N\,P \text{ .... (X.)}$$

Care must be taken to get the zero of the brake accurately. For heavy brakes this can be done by standing on the scales, and with the brake perfectly loose note the scale readings as the brake arm is slowly raised, and again as it is slowly lowered. Our zero will be

$$Z = \frac{(m - w) + (m' - w)}{2} \text{ .... (XI.)}$$

m = scale readings up, m' = scale readings down, and w = weight of man.

In Fig. (V.) is shown one form of a rope brake. With this brake the scales can be easily balanced at the beginning of the work, and the net scale reading can then be taken at once. The results are worked up as with the Prony brake. The heat can be conducted away as with the Prony brake, while a small amount of water can be sprinkled on the rope from time to time to keep it from charring. For large powers more than two turns of rope can be used. This form of brake is easily constructed, and for powers of 50 horse or less works very uniformly and satisfactorily.

Where the engine is non-condensing a separating calorimeter can be placed in the cylinder exhaust pipe, and having the quality of the steam entering and leaving the cylinder and the heat equivalent of the work done, we can easily find the radiation loss of the engine.

Preliminary Precautions and Final Results.

After the apparatus is ready for use a short preliminary run should be made to see that everything is working correctly. Special care should be taken to see that the valves, piston, drains, air pump or condenser do not leak, while the indicators, thermometers and gauges should be calibrated immediately after the test. Some form of log blank for original data and for final results should be used, as shown, complete with two sets of figures at the end of this article.

The friction horse power equals

$$\text{F.HP.} = \text{I.HP.} - \text{D.HP.} \text{ .... (XII).}$$

The mechanical efficiency equals

$$\text{Mechanical efficiency} = \frac{\text{D. H. P.}}{\text{I. H. P.}} \text{ .... (XII.)}$$

The mech. eff. will increase with the load up to the power of the engine. In other words, the power required to drive the engine itself is nearly constant at all loads.

The heat lost by radiation per 100 strokes (*H*) can be found as follows:

$$H = M\,(x\,r + q) - \left(M\,(x'\,r' + q') + \frac{p\,l\,A\,N'}{3778}\right) \text{.... (XIII.)}$$

Where M (x r + q) = the heat entering the cylinder.
M (x' r' + q') = the heat leaving the cylinder.
M = weight of steam per 100 strokes.
$\frac{p\,l\,A\,N'}{778}$ = work done in B.T.U. per 100 strokes.

(*x*) being the quality of the steam, (*r*) the latent heat and (*q*) the heat of the liquid. (See article on Calorimetry, April number of MACHINERY.)

Log of Simple Engine Test.

Engine built by .... Tested by .... Date ....
Diam of cyl., 12 inches. Stroke, 36 inches. Diam. piston rod, 2 inches. Length of brake arm, 5 feet 6 inches. Tare of the brake, 90 pounds. Scale of ind. spring, 60 pounds. Barometer, 14 pounds.

No.	Time.	Revs per minute.	Gauge Pressures.				Temp't's.		Weights, Pounds.				
			Steam Pipe.	Steam Chest.	Exhaust.	Calorimeter.	Calorimeter.	Engine Room.	Boiler Feed or Steam.	Brake.	Cond. Steam Sep. Cal.	Water Sep. Cal.	Condensing Water.
0	2.	90	115	110	1	1	270	......	0	......	......	....	....
1	2.10	90	116	111	1	1	270	75	295	745	1.2	2.4	....

Result Blank–Simple Engine Test.

Average weight of steam per 100 strokes.............. 16.35 lbs.
Actual water rate per I.HP. per hour................ 26.08 lbs.
Water rate per I.HP. per hour, corrected for cal...... 25.58 lbs.
Condensing water per lb. of steam........lbs........ lbs.
Heat entering cyl. per 100 strokes at 129 lbs...... 19,202 B.T.U.

Heat equivalent of work per 100 strokes............ 1,570 B.I.U.
Heat discharged per 100 strokes..................16,116 B.I.U.
Heat lost by radiation per 100 strokes............ 1,516 B.I.U.

No.	M. E. P.		I. H. P.		Total H. P.	D. H. P.	Mech. Effi. %.	Absolute Pressures.			Quality of Steam, %.	
	Head.	Crank.	Head.	Crank.				Steam Pipe	Exhaust.	Cal.	Ent.	Leav.
0	38	36	35.2	33.3	68.5	61.7	90	129	15	15	98.5	83.3
1	39	36	36.1	33.3	69.4	61.7	90	130	15	15	98 5	83.3

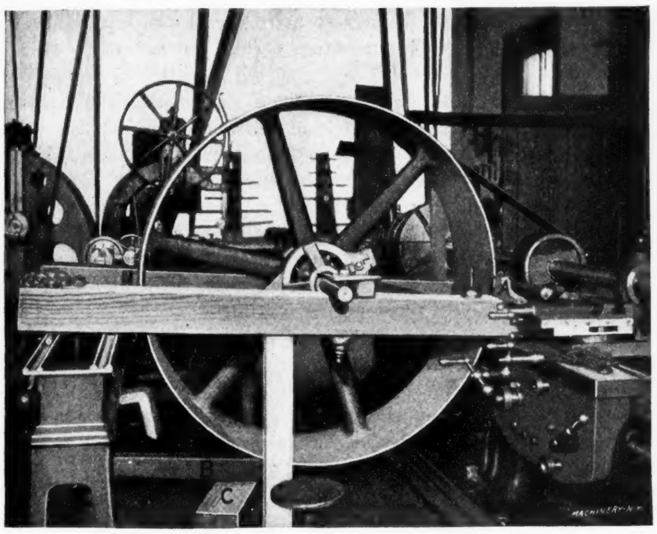
* * *

A LARGE JOB UNDER DIFFICULTIES.

Wm. F. Long.

The acompanying photograph is of a device the writer built for turning and boring a pulley 74 inches in diameter and 21 inches face. We call ourselves "The Up-to-Date" machine works, and such we are up to a certain limit. Being proud of the title we do not like to let anything go by, and often devise such things as shown in the illustration to overcome difficulties that come in our way. As will be seen by the photograph, a 24-inch Putnam and 17-inch Lodge & Shipley lathe were used in the operation of turning the pulley, by bolting two 6 x 6 timbers between them.

The pulley was centered by a spider on either side of the hole, the same carrying a star-feed boring bar. The bar being level and trammed from the large lathe, it was securely bolted to the timbers, and then the small lathe was leveled and trammed to the large lathe and spikes were driven in the floor at the outer side of the legs. The lathe was then set over to give the right crown to the pulley. This moved one of the lathe legs away from its spike and a block was placed between the spike and the leg to keep the distance, after which the bolts through the timber were securely fastened.



MACHINING A LARGE PULLEY WITH SMALL TOOLS.

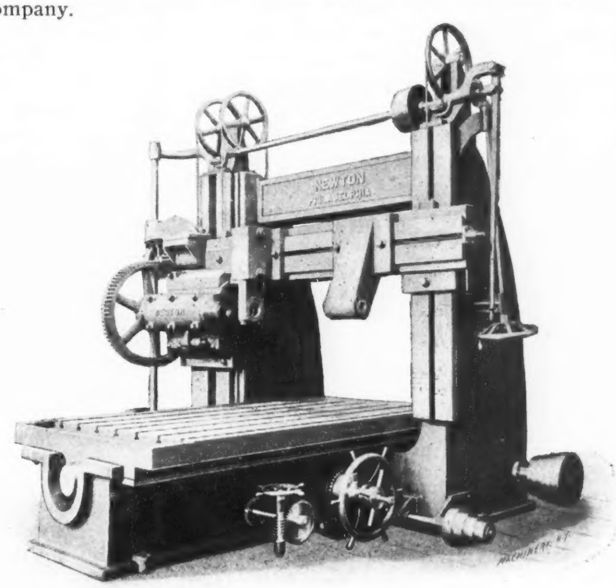
As will be noticed, the large lathe has a shafting rest in place, which supports a piece of shafting held between the centers and driven from the face-plate of the lathe. On this shaft we put a 12 inch x 6-inch wood pulley, and geared the lathe to the proper speed to turn 12 inches in diameter, and by belting to the 74-inch pulley secured the proper speed for turning this diameter. An idler was used on a shaft between the two pulleys, which kept the belt at a good tension.

The belt having been arranged, a tool was bolted to the carriage of the small lathe and the cut started. The pulley was so limber, however, that it chattered very badly, and this was overcome by placing a roller on either side of the rim, as at *A*. The play in the arbor was overcome by placing a roller on timber *B*, using *C* as a fulcrum and placing weights on the outer end of *B*.

The feed for the carriage of the small lathe was obtained by belting over from the pulley shaft in the large lathe. After the pulley was turned, which required 11½ hours, with a maximum cut of ¼ inch, the pulley was lowered to the floor and bored to size with the boring bar referred to, making as presentable a job as could have been done on some of our fine mills.

A LARGE HORIZONTAL MILLING MACHINE.

The accompanying illustration is of a large horizontal milling machine made by the Newton Machine Tool Works, of Philadelphia, Pa. It is, in fact, believed to be the largest standard milling machine made in this country. It is not an enlarged machine, but is one of the standard types regularly made by this company.



A MASSIVE MILLING MACHINE.

It weighs 66,000 pounds, complete, and will take work 7 feet wide between the housings. The carriage is 6 feet wide, 10 feet long, and the center of the spindle can be raised 60 inches from the carriage. The spindle is unusually stiff, being 7½ inches in diameter, and has cross adjustment for convenience in setting the cutters to the work. The rail has a central support for the cutter arbor, which can be used where there is a gang of cutters whose collars will admit the use of the support. The massive spindle head is counter-weighted and has a rapid adjustment by power. The carriage has eight changes of automatic feed, with a quick movement by power in either direction. This movement of the carriage is controlled by a small hand wheel on the working side of the machine, operating the friction on the opposite side of the machine, as shown in the illustration.

* * *

JAMES WATT.

The work done by James Watt in the line of steam engineering was a credit to him and an honor to the age in which he lived. He not only invented the double-acting engine, means for using steam expansively, the condenser, the steam jacket, the steam engine indicator, the governor, and first applied the crank to convert reciprocating motion into rotary motion (although the idea was subsequently stolen from him), but some of his engines have stood the test of time in a remarkable manner. A striking illustration of this is an engine constructed by Boulton & Watt in the year 1777, which has been in continual service in a pumping station at Smethwick, England, from that date until the present year, a period of 121 years. It has now been placed out of commission, and whether or not it is in the condition of the Dutchman's trousers, "all patch and no breeches," we do not know, but it hardly seems possible that much of the original machine can remain.

This performance is all the more creditable to Watt, because the workmanship upon his engines was immeasurably crude compared with the workmanship of to-day; in fact, we remember to have seen stated by some biographer that Watt made the claim that one cylinder and piston that he had completed was finished so accurately that it was impossible to pass a coin larger than a certain small English piece (about about as large as a 10-cent piece), between the piston and the walls of the cylinder.

Watt died in 1819, and it is a significant fact that from his time to 1849, when Corliss introduced his engines, there were no marked advances in stationary engineering practice. The compound engine had been brought into prominence by Woolf fifteen years before his death, and while great strides were made in locomotive and marine engineering during the first half of the century, the stationary engine was too well developed at the beginning of the century to permit of much improvement with the materials then in use and the methods that were then in vogue.

SHOP TALKS WITH YOUNG MECHANICS.—8.

CALIPERS.

W. H. Van Dervoort.

Calipering tools may be classed under two heads—transfer and recording. The transfer class includes all calipers used to transfer scale dimensions to the work, work dimensions to the scale or to receive and transfer from one piece of work to another a dimension without reference to its exact magnitude. The recording caliper is provided with a scale which gives the length of the dimension measured as well as forming a transfer instrument. This latter class includes the vernier and micrometer calipers. The large and heavy calipers of this class are called "bench micrometers." Recording calipers are usually graduated to read to one-thousandth of an inch, which makes them sufficiently accurate for use on all ordinary mechanical operations. When, however, the tool is designed for greater accuracy, it becomes necessarily more delicate and expensive, and is not suitable for general shop work. Such instruments come under the class known as "measuring machines," which are used as test instruments.

The transfer calipers are of three general forms—firm joint, lock joint and spring—the latter two usually being called adjustable calipers.

Calipers to be first-class should have well proportioned legs made of high-grade steel, preferably tempered, with a carefully fitted joint having a uniform amount of friction at all positions of the legs, thus insuring a smooth, steady motion when setting.

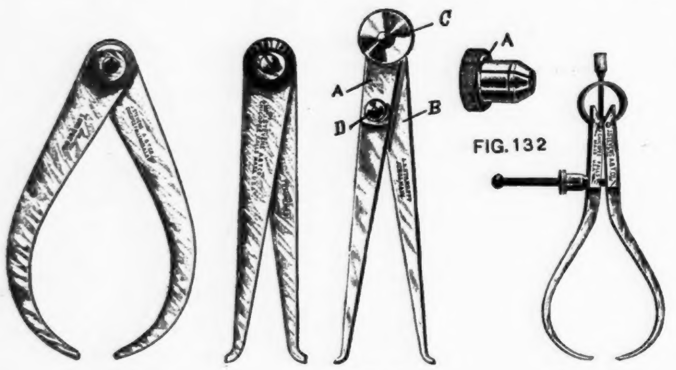


FIG.129 FIG.130 FIG.131

In Fig. 129 are shown a pair each of inside and outside firm joint calipers. Fig. 130 shows a pair of inside lock joint calipers. In this tool a tapered socket joint enables the short arm *A* to be rigidly locked with the outside leg *B* by tightening the knurled nut *C*. The nut *D*, which is coned on the bottom, moves over a stud, which is secured in the middle leg, and moves through a slot in *A*. Secured to *A* is a small cone, against which the cone nut bears, thus forcing the middle leg away from *A* when *D* is tightened down. A stiff spring set in the under side of *A* resists the separation and holds the joint steady. This tool may be set to approximately the size desired, the joint locked and the final adjustment obtained by turning the nut *D*. By adjusting *C* to give the correct friction this tool may be used as a firm joint caliper.

In Fig. 131 is shown a good example of the numerous forms of spring calipers in use. Its construction is clearly shown in the figure. They may be had with solid or slip nut on the screw, a slip nut being shown in Fig. 132. It is a split nut pivoted at *A* with a light spring in a recess inside the knurled head, which holds the halves together at the threaded end, the outside of which is coned to fit a recess in the post or caliper leg, which prevents the nut from opening when the tension of the joint spring is upon it. The slip nut saves time in setting the calipers, which is of material importance where numerous settings are to be made.

The skillful use of transfer calipers depends entirely on the good judgment and delicate touch of the operator. He must recognize contact, between the points of the caliper and the work, without pressure. The ability to make a sure calipered fit is an accomplishment that comes only with practice. In setting calipers to a scale care must be exercised, the chance for personal error being great. In setting the outside calipers the scale should be held vertically in the left hand, with the end of the little finger resting against the side at the bottom end to steady the lower point of the calipers over the end of the scale, the caliper being held in the right hand, as shown in Fig. 133. If the caliper be a "firm joint" it must be adjusted to the required dimension by tapping the legs against some solid body, the force of the blow diminishing until the proper adjustment is obtained. In setting the adjustable calipers, they should be held in the right hand, as shown in Fig. 134, the thumb and fore-

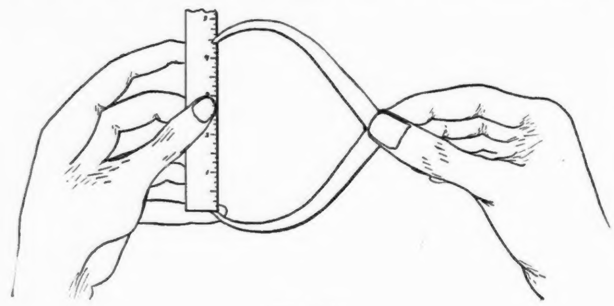


FIG. 133.

finger operating the adjusting nut. The upper caliper point rests against the side of the scale and over the graduations. The lower point rests against the end of the scale, but the eye must determine when the upper point coincides exactly with the required reading. The end of the scale should be square, only a comparatively new steel scale being suitable, as an old one is frequently worn enough to make it appreciably short. When the caliper is heavy the little finger of the right hand should be pressed against the lower leg to steady and support it. The slight angularity of the points with the plane of the scale, due to dropping the lower point below the graduated surface, has an appreciable effect on the accuracy of the setting only when adjusting for the smaller dimensions. This error may be avoided by slightly twisting the caliper so as to spring the legs sidewise enough to bring the points in planes parallel to the surface of the scale.

In setting the inside calipers to the scale the end of the scale should be placed against and held at right angles to a plane surface; one point of the caliper placed against this surface and the other adjusted to the required reading on the scale. It is very important that the scale be held perfectly at right angles, as a slight variation makes considerable error in the reading. It is not good practice to set both points to lines on the scale, as the chance for error is much greater than when the measurement is referred to one end of the scale.

In transferring a setting from one pair of calipers to another, the reference pair should be held in the left hand, with the lower point resting against the end of the little finger. The lower point of the pair being set should be brought in contact with this point, the end of the finger serving to steady the points.

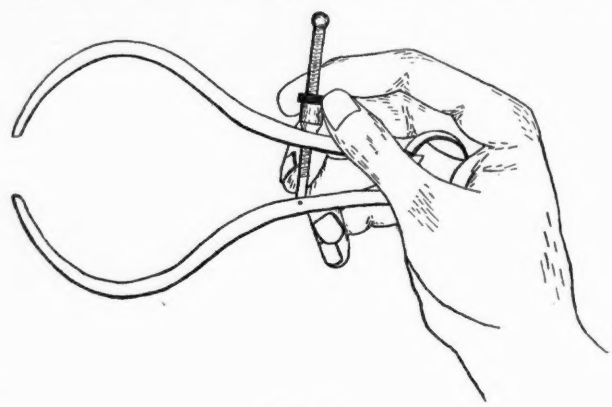


FIG. 134.

The upper point should then be so adjusted that it just makes contact with the upper point of the reference pair. The accuracy attained depends entirely upon the skill with which this adjustment is made, other than the slightest pressure between the points serving to distort the measurement.

Assume that a shaft is to be turned to 4⅜ inches in diameter and a hub bored to fit this shaft. The outside caliper will be set to size, as shown in Fig. 133, involving the first chance for error.

The bar will now be turned and finished to the caliper size, allowing a second chance for error. The third opportunity for error arises in the setting of the inside to the outside calipers, and finally the fourth in boring the hub to the dimension of the inside caliper. Of these the first two affect the diameter of the work and the last two the fit.

In using the outside calipers they must pass squarely over the work, just touching it. The lighter the touch the greater the accuracy attainable. For final calipering the work must be stationary, as a rotating piece will draw the calipers over long before it is down to size. When a number of pieces are to be turned to sample, it is permissible to set the calipers so that the contact will just sustain the weight of the tool, turning all of the pieces so the contact will be the same. This, however, can hardly be called calipering.

In setting the inside calipers to the outside, the contact between points should be, as near as a delicate touch can determine, the same as the contact between outside caliper points and the work. The work should then be bored to that diameter which gives the same contact between the inside calipers and the walls of the hole as between the inside and outside calipers.

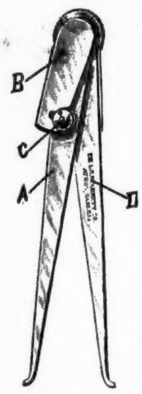


FIG. 135


FIG. 136


FIG. 137


FIG. 138

The caliper shown in Fig. 135 is technically known as a "transfer" caliper. Frequently it is necessary to caliper a piece of work the shape of which makes it impossible to remove the calipers without loosing the setting. In this tool the leg *A* is secured to the blade *B* by the stud and nut *C*, the stud striking against the end of the small slot. The caliper is then set, for example, to the larger diameter of a chambered cylinder, and the joint nut tightened, securing *D* and *B* to each other. C can now be loosened and the leg *A* moved to allow the tool to be removed from the work. When moved back against its stop it will show the original setting. Only calipers of the very best construction should be depended upon for this kind of work.

The thread caliper is shown in Fig. 136. It has wide, thin points for calipering both the top and bottom of screw threads.

Fig. 137 shows a hermaphrodite caliper, a tool having one caliper and one divider leg. The divider leg is preferably adjustable, and when caliper is of the spring pattern the caliper leg must have right and left feet, as the legs cannot pass by each other as with the firm or lock joint patterns. This caliper is a valuable tool for centering, and scribing purposes.

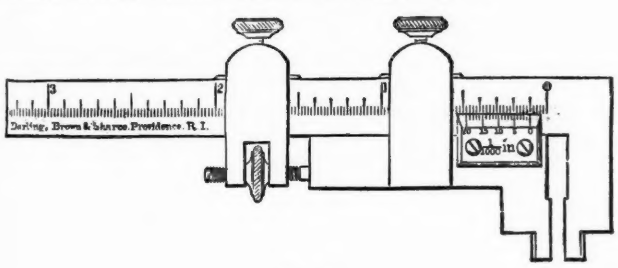


FIG. 139.

The key hole caliper, shown in Fig. 138, is a valuable tool on work the shape or position of which will not allow the use of two curved legs.

The vernier caliper, an example of which is shown in Fig. 139, is a form of beam caliper which depends for its setting on a graduated scale on the side of the beam, the subdivision of the graduations on the beam being obtained by means of the vernier on the sliding jaws. This tool possesses the advantage of wide range and comparatively light weight. Its value as a measuring instrument depends on its accuracy of construction, cheap calipers of this class being of little value. The jaws should be of tool steel, carefully tempered, and the faces ground exactly parallel with each other. The sliding jaw is given its final adjustment by means of the small knurled nut and screw in the auxiliary slide. The graduations usually read to thousandths on the front side of the beam, and to sixty-fourths on the back. They may be used for either inside or out-

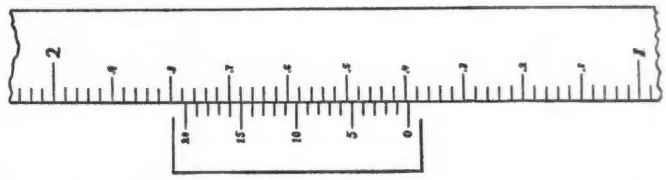


FIG. 140.

side measurements, but for inside readings on the vernier side a constant equal to the space occupied by the points must be added. On the back side of the instrument two zero lines are used, one for inside and one for outside measurements, thus avoiding the necessity of the addition for the inside readings.

In explanation of the use of the vernier: Referring to Fig. 139 it will be noted that the scale on the beam is divided into inches, tenths and fiftieths, and that the twenty divisions on the vernier cover exactly nineteen divisions on the scale. This makes each division on the vernier 1-20 of 1-50 less than a division on the scale, or 1-1000 of an inch. In the figure the zero on the vernier stands at .1 inch; if, now, the sliding jaw was moved out until the 10 on the vernier coincided with the next line on the scale (.3 inch) it would have moved through 10-20 of 1-50 = 1-100 or .11 inch from zero; and in like manner if the fourteenth division on the vernier had moved to coincide with the third line beyond 1.6 inches, as shown in Fig. 140, the reading would be 1 inch + .3 inch + 4-50 (.08) + .014 = 1.394 inches.

Other graduations than the one shown may be used; for example, if the beam is graduated into forty lines per inch and the vernier has twenty-five divisions covering twenty-four on the scale, then each division will be 1-25 shorter than one on the scale and 1-25 of 1-40 = 1-1000 of an inch.

FIG. 141.

A magnifying glass is of great value in reading these instruments, especially so when many readings are to be made in a short space of time, it being very trying on the eyes.

The micrometer caliper, a skeleton view of which is shown in Fig. 141, is a tool that has come into very general use among mechanics employed on work requiring uniformity and accuracy. Its simplicity and relatively low cost makes it an instrument that even the mechanic on comparatively coarse work can scarcely afford not to have in the caliper drawer of his tool chest.

These calipers may be divided into two classes, the yoke and the beam patterns, Fig. 141, illustrating the former. In this figure the construction is quite clearly shown. The shank of the yoke contains at its outer end a split nut which, for adjustment for wear, may be closed onto the screw by advancing the nut toward the yoke, the stem being threaded on a slight taper. The graduated shell or thimble is attached to the end of the screw and rotates with it, moving along and over a shell on the shank. As the screw does not extend out of the yoke it is completely encased and protected from dust or injury for all positions of the measuring point within the range of the instrument. The small knurled extension to the thimble serves as a speeder for rapidly advancing the screw. The knurled nut in the yoke contracts a split bushing over the measuring stem, thus locking it in any desired position. The measuring point and the anvil against which

it strikes are carefully hardened and ground, making the surfaces parallel planes.

The micrometers of this class are usually provided with a screw having forty threads per inch. The barrel is graduated to tenths and fortieths of an inch. Thus one revolution of the screw advances the thimble one division on the barrel, which equals 1-40 of an inch of .025 inch. The circumference of the thimble is divided into twenty-five equal parts, then 1-25 of one revolution of the screw advances the measuring point 1-25 of 1-40 = 1-1000 of an inch. When the end of the measuring point

FIG.142

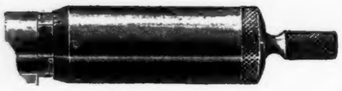


FIG.143

and the anvil come in contact, the zero points on barrel and thimble should coincide, thus avoiding the necessity of a correction for each reading. As the measuring faces wear slightly, it is necessary to provide some means of adjustment in order to keep the zero reading correct. In this particular instrument the barrel may be rotated on the shank sufficient to bring the lines correct. In others the anvil may be forced ahead by means of a small screw in the yoke.

As these instruments are graduated to read decimally, a table of decimal equivalents is usually stamped on the sides of the yoke, eighths, sixteenths and thirty-seconds on one side, and sixty-fourths on the opposite.

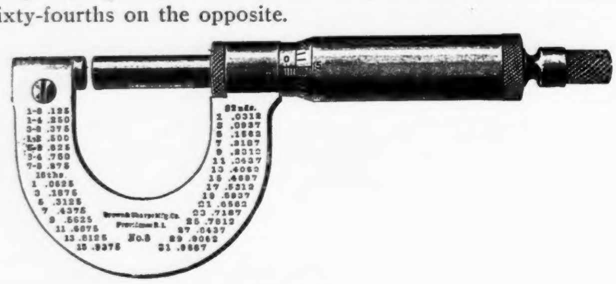


FIG. 144.

In Fig. 142 is shown a micrometer caliper to take sizes between one and two inches. When the zero points coincide, the face of the measuring point is exactly one inch from the face of the anvil. A 1-inch reference disc, hardened, ground and lapped to size is furnished with each caliper with which to test the correctness of the setting.

Although a half, or even a fourth of one-thousandth can be quite readily approximated on the reading, which brings it within the accuracy limit of all ordinary work, it is frequently desirable to get at these fine readings more closely. For this purpose the vernier is applied to the barrel, as shown in Fig. 143. It con-

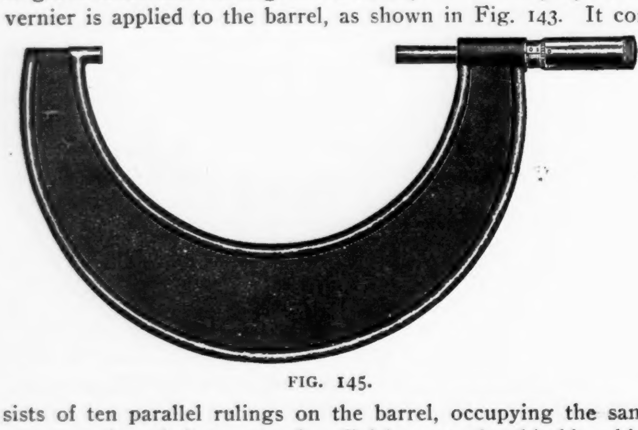


FIG. 145.

sists of ten parallel rulings on the barrel, occupying the same space as nine of the twenty-five divisions on the thimble which makes the spaces on the barrel one-tenth shorter than that on the thimble, thus giving 1-10 of 1-25 of 1-40 = 1-10,000 of an inch. This refinement can be relied upon only when the greatest care is exercised in making the measurement, as the slightest excess of pressure on the screw over that at which the caliper is adjusted will spring the instrument more than the minute distance it is expected to indicate.

In the use of the micrometer caliper it is important that the pressure of the measuring point against the work is the same as the pressure of the point against the anvil when the zero setting is made, as it is quite possible for a careless workman to force the screw enough to set the thimble zero two or three thousandths past its zero position. For all ordinary work the following method will serve well: Adjust the caliper so the zeroes will coincide when the thimble is turned with just enough pressure to raise the yoke to a horizontal position. In calipering, hold the work in a vertical position, and with the caliper in the right hand adjust the measuring points onto the work until the yoke again comes up to the horizontal position, thus insuring practically the same pressure between point and work as between point and anvil when zero setting was made.

For very fine work the application of the friction drive to the thimble is an advantage. In Fig. 144 is a micrometer having such a device. The knurled extension contains a ratchet which, when the pressure reaches the desired point, slips. In backing the screw the ratchet engages the pawl, making a positive drive. The advantage of the ratchet over a plain friction is that the screw can be backed more rapidly without slipping.

FIG. 146.

Micrometer calipers of greater capacity than 2 inches have until recently, been little used. The introduction of the reliable, moderate-priced caliper, shown in Fig. 145, has met a popular demand for a yoke caliper of large size. It is a much more convenient tool for the workman than fixed caliper gauges. The yoke section is a bulbed I, which gives light weight strength and an excellent grip for the fingers. All adjustments for wear are in the head. For the table of decimal equivalents usually stamped on the yoke the following is substituted: Every fifth graduation on the barrel (.125 inch) is extended, and beginning

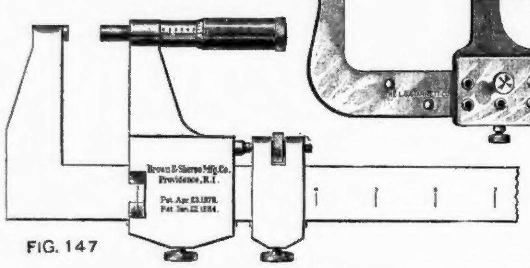


FIG. 147

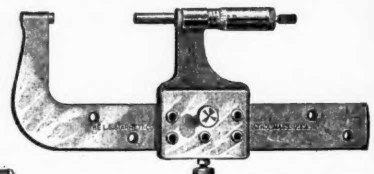


FIG. 148

at the zero marked 1 to 8, inclusive, thus giving an inch graduation by eighths. On the thimble are stamped the decimal equivalents for 1-16 inch, 1-32 inch and 1-64 inch, thus giving a contracted conversion table, the application of which requires at most only a simple calculation, thus, 23-32" = ⅝" + 1-16" + 1-32". Set to the 5 and add the sum of the decimal equivalents for 1-16 inch (.0625 inch) and 1-32 inch (.0312 inch) = .0937 inch, which should be set back from the ⅝ in the ordinary manner. This does not interfere with reading the caliper decimally in the ordinary way.

This caliper is made in six sizes, from 1 to 6 inches. The 1-inch, 2-inch and 3-inch sizes have yokes made from drop forgings of bar steel, and the 4-inch, 5-inch and 6-inch yokes of steel castings, all neatly finished and japanned. The face of the anvil is formed by a hardened steel plug of same diameter as the end of the measuring point.

The measuring range of each size is 1 inch, and for adjustment of all sizes other than the 1-inch caliper, standard end measure test pieces are required.

In Fig. 146 is shown a form of yoke caliper but recently placed on the market. It is made in two sizes, one of which measures from zero to 3 inches, and the other from 3 to 6

inches. The anvil is mounted in the end of a spindle, which is provided with stops exactly 1 inch apart. A slight turn of the anvil spindle when either stop is to be used brings it firmly against its seat, in which position it is securely clamped.

Beam micrometer calipers are illustrated in Figs. 147, 148 and 149; in Fig. 147 the inch settings are made to accurately graduated rulings on the beam.

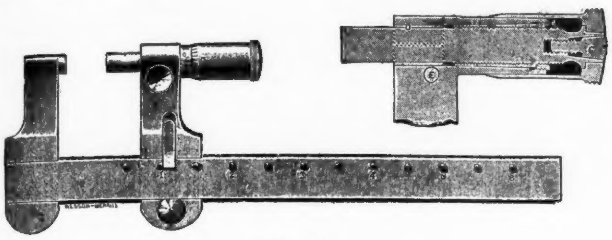


FIG. 149.

In Fig. 148 these settings are made by inserting the tapered steel pin in the holes in the sliding jaw and their corresponding holes in the beam. A separate set of holes is used for each setting. The holes are bushed with hardened steel bushings ground and lapped to fit the tapering plug. The stops on the caliper shown in Fig. 149 are hardened steel pins set in the beam and against which the finger on the sliding jaw makes contact. In this instrument the range of the measuring point is ½ inch, the stops being placed that distance apart.

The capacity of the beam micrometer for measuring flat work is limited only by the length of the beam. For round work the height of the jaws limits the diameter, usually to about 4 inches, since in order to keep the weight of the instrument within reasonable limits it is not advisable to make the jaws much greater than 2 inches high.

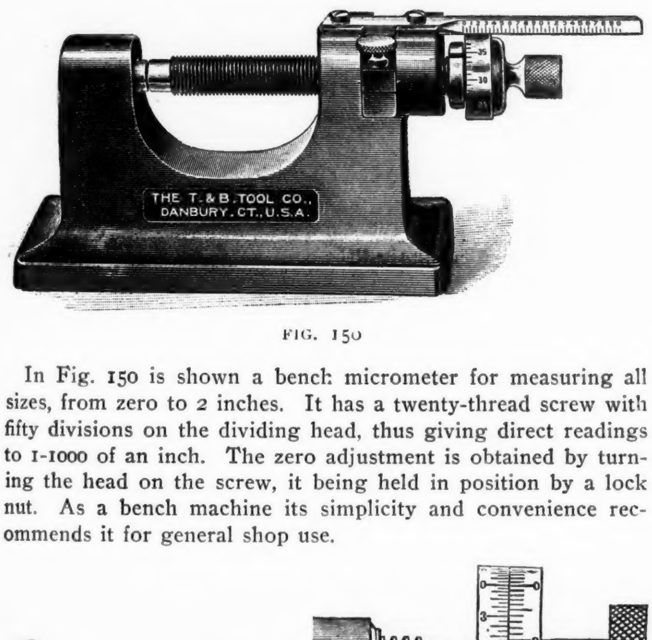


FIG. 150

In Fig. 150 is shown a bench micrometer for measuring all sizes, from zero to 2 inches. It has a twenty-thread screw with fifty divisions on the dividing head, thus giving direct readings to 1-1000 of an inch. The zero adjustment is obtained by turning the head on the screw, it being held in position by a lock nut. As a bench machine its simplicity and convenience recommends it for general shop use.

FIG. 151

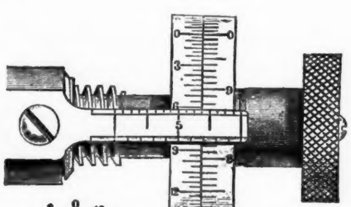


FIG. 152

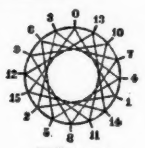


FIG. 153

Fig. 151 shows the Sweet's measuring machine, which is made in 4, 6 and 8 inch sizes, and may be classed under the head of bench micrometers. This instrument, which is intended for practical shop uses, reads as regularly furnished to 1-1000 and 1-1280 of an inch. When required a vernier is used on the head, which gives readings of 1-10,000 of an inch. The range of measuring screw is 1 inch, test pieces being furnished for setting the sliding anvil to the inch zero positions.

Fig. 152 shows a portion of the measuring head. A 1-10 inch pitch trapezoidal thread measuring screw is employed. This form of thread gives a square bearing on its work side, and the quick pitch facilitates rapid adjustment. The knurled thumb nut drives through a friction. The outer disc of the dial is divided to hundredths, thus giving for each division 1-100 of 1-10 = 1-1000 of an inch. The reading is made on the front edge of the index bar.

For the fractional readings the left-hand disc is used. It is divided into 128 parts, and every eighth division numbered, as shown in Fig. 153. One revolution of the screw equals 1-10 of an inch = 128 divisions on the dial, whence one division = 1-10 of 1-128 = 1-1280 of an inch. The upper edge of the index bar is graduated to sixteenths for convenience in getting the approximate setting. All readings are, however, made on the front edge of the bar.

Referring to Fig. 153: Following the straight lines from 0 to 1, 2, 3, etc., back to zero (16) five complete revolutions are made, which carries the screw back ½ inch. Then every five divisions are 5-16 of 1-10 = 1-32 of an inch at the measuring point, and every 2½ divisions = 1-64 of an inch; and since each division is divided into eight equal parts on the disc, 1-128, 1-256 and

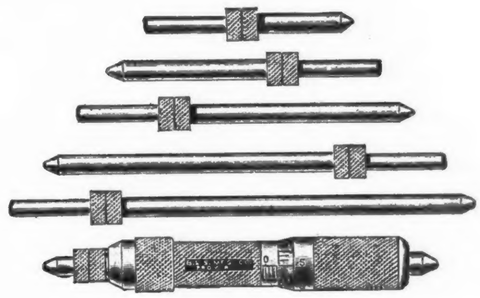


FIG. 154.

1-1280 may be found by using 10, 5 and 1 of these small divisions, respectively; for example, in Fig. 152 the reading line is 6-32 inch beyond the 16-32 (.5) inch mark = 22-32, and the 6 on the disc should nearly coincide with the 22-32 division. Bring the 6 to read at the lower edge, and the measuring point will be 22-32 inch from the anvil. If 23-32 inch had been wanted, the 7 on the disc would have been brought around to the reading edge. If 47-64 inch was required, the 7 would be carried past the reading edge twenty divisions, and in like manner ten more divisions would make it read 95-128 inch.

Any error in the pitch of the measuring screw which would affect the number of turns per inch is corrected by setting the index bar at an angle with the axis of the screw. Assume, for example, that the ten turns of the screw advances the measuring point 1-1000 inch too far. The screw is too long, and less than ten revolutions should be made by an amount equal to one of the divisions on the outer disc. The outer end of the index finger will be raised this amount above the inner end graduation. This corrects the error proportionately from end to end of the screw.

An example of the inside micrometer caliper is shown in Fig. 154. It is used for making inside measurements and reads to thousandths through a ½-inch range. The instrument shown with its extension rods makes any measurement between 3 and 6 inches. The nut and check nut on the extension rods may be adjusted down to compensate for any wear of the points.

* * *

ABOUT "HOW AND WHY."

In answer to the third question in the "How and Why" department in the September issue, I will say that I use ammonia, such as is put up for household use, and wipe off with kerosene to prevent rust. F. J. PERRY.

Lawrence, Mass.

If your correspondent J. E. M., in the September issue, who is bothered with lime in the ports of his pump, will drain the water out and remove the valves, then pour the ports full of muriatic acid and let it stand over night, he can wash the lime out with a hose or scratch it loose with a wire. He could fit a plug of wood in the cylinder so it would not take so much acid; I think undiluted acid is best and does not cost much.

ARCADE.

WHAT MECHANICS THINK.

A DEPARTMENT FOR THE EXPRESSION OF PRACTICAL IDEAS OF INTEREST, TECHNICAL OR OTHERWISE.

Write on one side of the paper only, and when sketches are necessary, send them. No matter how rough the sketches may be, we will see that they are properly reproduced.

A METHOD FOR MAKING NUMBERING PLATES.

A cheap and satisfactory method for making numbering plates for tool racks, compartments in drill and wire drawers, etc., is to cast them in soft metal in metal molds, as shown in the accompanying sketch.

A and *B* are pieces of metal, preferably brass, each smoothed off on one side. *A* has a depression milled into it the shape the plate is to be cast. *B* has two holes, *e* and *f*, drilled through it, into which bushings like *g* are driven. Each bushing has a hole, *i*, drilled through the center for holding brads or tacks loosely. There is a small projection, *h*, on the inner end of *g* for the purpose of holding up the head of the tack away from the bushing, so as to allow the hot metal to flow all around it. The bushings are used for the purpose of getting these projections, *h*, close to the tack, which could not be done conveniently if no bushing were used.

The two pieces, *A* and *B*, are held in proper relation to each other by dowel pins, which are driven in one piece and are loose in the other. For convenience two pieces, *m m*, may be tapped into *A* and *B*, to serve as handles, the lower one of which can be held in the vise, if desired.

Fig. 1 shows one style of plate that is very satisfactory. Fig. 3 is a small plate handy for drill or drill-rod drawers. These plates may be stamped with large figures, and then filled with black paint or varnish, which makes them very conspicuous.

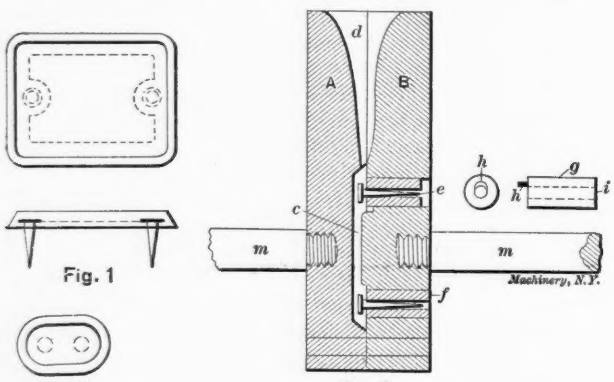


Fig. 1 Fig. 3 Fig. 2

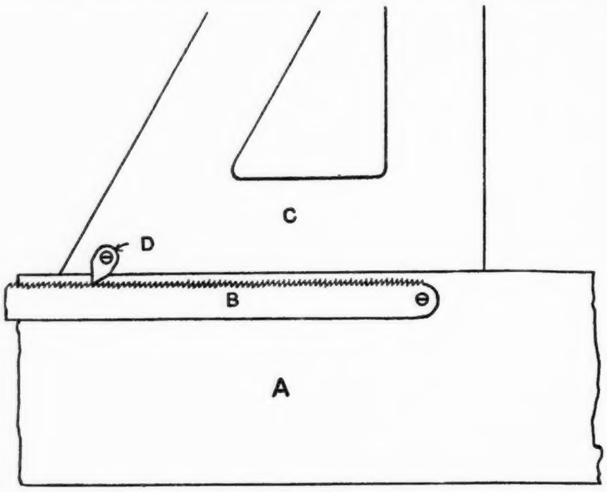
I have found soft metal castings very convenient in making models, where there are a number of pieces alike. Metal cores may be used and pieces produced that it would not be practicable to make in any other way. Britannia metal, Babbitt, tin, lead or solder may be used. The mold should be kept hot, which will be the case, of course, if the castings are made steadily.

Meriden, Conn. JAMES P. HAYES.

(Mr. Hayes has sent us a sample name plate, which is the neatest device for the purpose intended that we have seen. The casting is smooth, the brads are held firmly in place, and the figures, which have evidently been stamped upon it and filled with black varnish, are legible from a long distance. He also sends some small castings of lead or similar material, used in a model of a shaking grate. There were about 400 of these pieces to make in all, and they required a somewhat complicated core. They are nicely cast, however, and are interesting specimens of work.—Editor.)

A SIMPLE SECTION LINER FOR DRAUGHTSMEN.

Having experimented with several section liners, and finding them all of little use, I hit upon the following plan which is simple and practical: *A* is a straight edge on a T-square blade, and to it, and near one edge, is screwed a hack-saw *B*; on a triangle *C* near one corner is fastened a small piece, *D*. By moving the triangle toward the right, the point of the brass pawl is lifted over the edge of the teeth of the hack-saw, and by a slight movement to the left is brought up against the square face of the teeth, which gives the necessary spacing. By very little practice the lines may be drawn very rapidly, and all the same distances apart, as there is very little variation in the distance between the teeth. By putting the pawl *D* near one of the square corners, lines about eighteen to the inch may be drawn; by placing at the 60-degree angle, as shown in the cut, there will be about twenty-one lines to the inch; by placing at the 45-degree angle,



there will be about twenty-five lines to the inch; by placing at the 30-degree angle there will be about thirty-six lines to the inch, and by turning the hack-saw blade over, and placing the pawl on the opposite side, lines may be drawn from the right to the left if desired. W. B. RUGGLES.

New York, N. Y.

HE SMILED ("NIT").

Sadler was full of genius, perseverance and pluck. With him the lack of a suitable tool was no hindrance, but with true Yankee inventiveness would quickly devise some makeshift to fill its place. This faculty on one unhappy job, however, came near spoiling his reputation.

Having secured a steam fitting job for the residence of a wealthy railroad man, it was necessary to make a small copper pipe coil. As the work was being done on the grounds, no suitable mandrel was at hand, but Sadler's eagle eye quickly spied a shade tree of "near enough" dimensions on the lawn, and very soon a beautiful spiral had been carefully wound about its shapely trunk. Then for the first, Sadler realized that to secure his coil it was necessary to cut down the tree. Permission to do this was promptly refused, and with the atmosphere strongly charged with the odor of brimstone the coil was slowly and sadly unwound. F. E. R.

BORING CRANK-PIN HOLES.

For the benefit of D. D. F., "How and Why," No. 119, August issue, and others having similar trouble, I enclose description of a method for boring crank pin holes after the disc has been placed on the shaft and keyed, which I think has the merit of accuracy, and will save time over rigging up to drive a boring bar on a planer bed.

After boring the crank pin hole in the disc and mounting the disc on the shaft, mount the shaft and crank on the lathe carriage by blocking and bolting to it, and locate the position of the hole for the crank pin by placing a shaft or boring bar between the lathe centers and through the crank pin hole. The shaft may be moved to the back of the carriage if the swing of the lathe is not large enough to permit the shaft to be placed directly under the lathe centers. The shaft may now be aligned with the bar between the centers, and by having said bar of the same size as the shaft or with two collars slipped onto the bar of the size of the shaft, there is no calculation necessary to allow for difference in diameter of shaft and bar. Arrange a boring tool in the bar and use the carriage feed to feed the crank disc to the boring tool; or, if extra slide rest is at hand, it may be

bolted to the face plate and used, but where there is only, say 1-16 inch to remove from crank pin hole, the bar is simpler.

I have found this entirely satisfactory wherever used, and have used it myself. E. D. L.

A FORTY-FOOT CENTER INDICATOR.

Did you ever have to center a piece of work in a lathe, where the nicety of centering made it a tiresome and difficult matter to watch the end of the indicator? Well, here is my way to relieve the strain on your nerves, depending, however, upon the whim of Old Sol to assist us.

I assume that your indicating tool is one of those in which the indicating arm can be removed from the universal joint. Remove this arm and attach a small piece of plate mirror to it, one-quarter to one-half an inch square. This mirror is readily attached by the use of a little black wax procured at the shoe store. Attach it at such an angle as to project a beam of light to the other side of the room, and use a mirror in some nearby window to throw the ray of light upon the indicator mirror. Now, use that wabbling spot of sunlight on the side of the room for an indicator point, and be careful.

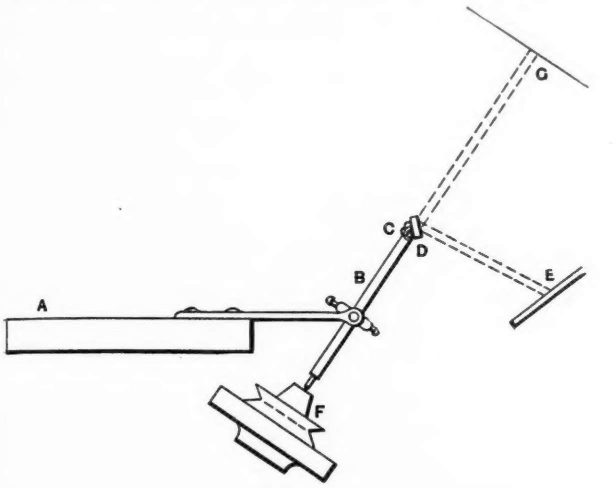


FIG. 1.

The arrangement is shown in Fig. 1, where *A* is the stock of the indicator, *B* the indicator bar in the universal joint, *C* the ball of black wax, *D* the plate mirror, *E* the looking glass at the window. The work being centered is at *F* and the reflection is thrown on the wall at *G*.

LATHE TOOL STOCK.

A convenient lathe tool stock can be made by using a piece of tool steel of a size adapted to your tool-post, as shown in the figure. Put the piece of steel in the shaper vise and plane out as indicated, leaving the section *A* not over 1-32 inch thick, and so that the square steel that you are to use will just clear and not bind on the top and bottom.

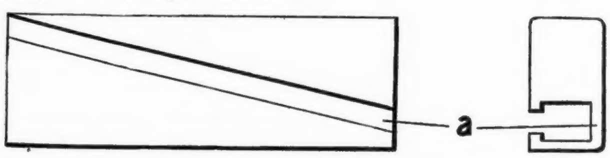


FIG. 2. LATHE TOOL STOCK.

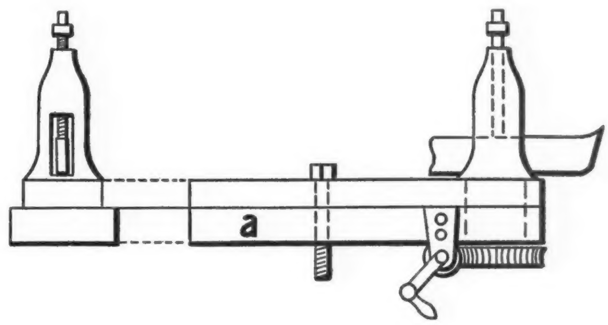
The steel can be a very free fit on the sides. Harden the stock and draw the temper, so that the thin rib *A* will be a good spring blue. The stock can easily be tightened upon the cutter steel by screwing down upon it in the tool-post so there will be no slip at all in use. H. O. W.

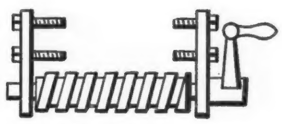
TURNING FILLETS AND PACKING RINGS.

The accompanying drawing is of an attachment for a lathe. It consists of a piece of cast iron, *a*, planed to fit in the tool post slot in the lathe carriage, where it is fastened. A capscrew passes through the block and is screwed into the carriage, allowing the block to project beyond it. In the projecting end is a reamed hole, say, about 2 inches in diameter; the tool-post is of iron or steel, the same as an ordinary tool-post, except that the lower end passes through the reamed hole and projects beyond on the bottom about 1 inch. A worm pinion is keyed on this end. Two pieces of steel are fastened by capscrews to the two sides of the block, and have holes near the bottom in which the axle of the worm runs. The pieces must be set close enough to enable the worm and pinion to mesh together. The worm is turned by a small crank or handle.

A tool fastened in the tool-post can be made to describe an arc of a circle (or a complete circle) by turning the handle, and different circumferences may be obtained by lengthening or shortening the distance from the tool post to the cutting edge of the tool.

This attachment is used for making fillets on connecting rods and will enable a man to make a fillet quicker, smoother and of better shape than by hand and with an ordinary tool-post.





The following is an excellent method for making packing rings:

Finish the outside of the casting first, and if the ring be eccentric, give it the required throw and bore out the inside and face off the edge. Measure off the required width of the ring and allow a scant sixty-fourth more for facing true. Take a cutting-off tool (bent or straight, if latter can be used), about 3-16 or 7-32 wide, and cut down until the iron on the inner side begins to bulge through. When it does that there is about 1-32 or little more of iron left. Withdraw the tool and move over so that the sharp corner takes a light cut down the side of the ring. Use a fine feed and a straight cut will be the result. Then take a side tool and face off the rest of the stock on the front side of the ring; then finish by using the cutting-off tool to detach the ring. While rings are being cut off, run a mandrel or centered piece of steel between centers to take up lost motion endwise. The result will be true rings. G.

DESIGN FOR SMALL CHUCK.

I have frequently had occasion to chuck small finished pieces in lathe or gear cutter, and I think obtained excellent results from the collet chuck herewith illustrated, in which *a* represents spindle of lathe or milling machine dividing head, *b* the hollow stem of chuck, *c* the collet, of which Fig. 2 is an end view; *d* is the knurled shell or nut.

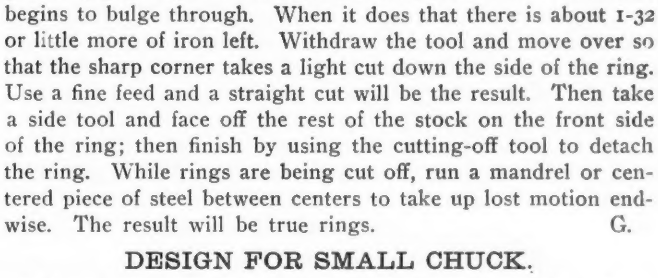


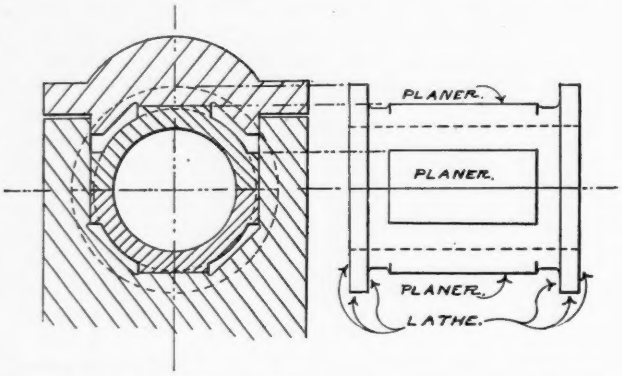
FIG. 1 FIG. 2

I find about 11 or 12 degrees for the back end and about 40 degrees for the front end will give a very strong positive grip to a finished piece when nut *d* is tightened with the hand. If collets are made to fit the work nicely, no trouble will be experienced either in grip or truth.

I have obtain excellent results with these chucks on automatic gear cutters, for holding small pinions with long stems, such as the first pinion in the apron of some lathes. C. F. P.

A PILLOW BLOCK BEARING.

I send herewith drawings of a bearing that I designed some three years ago. It has proven very satisfactory, and for that



reason I offer it to your readers. The construction of it and the time saved in the machine shop is readily seen by reference to the drawing. A. EDWARD RHODES.

BELTING AGAIN.

I have to thank Mr. Dunbar for his remarks upon the belting question in the last isse, in which he has some criticisms to make upon my article in the August issue; and my thanks are no less hearty because I disagree with him at almost every point. Every effort like his to lead young mechanics into the right path and to assist them in the journey onward is to be commended.

The first parting of the ways between Mr. Dunbar and myself comes with his statement that while the sum of the tensions in the tight and loose sides of a belt is a constant quantity whether the belt is at rest or is transmitting power; this is only true with vertical belts, and that in the case of horizontal belts the conditions are changed, owing to the effects of gravity. I am well aware that the horizontal belt will in general give less trouble in power transmission than vertical belts, because the effect of gravity is to increase the tension and to keep the belt tight in the former case, while in the latter case its whole effort goes toward making the belt slip on the lower pulley. But this has nothing to do with the relative tensions on the tight and loose sides of a horizontal belt. Gravity acts upon both sides of a horizontal belt equally, and can in no wise change the relations that exist between the tensions in these sides. At least, that is the way it appears to me.

If the tension on each side of a stationary belt is 100 pounds, and 50 pounds pull is required to start the pulley, then at the point of starting the tensions in the two sides will be 125 and 75 pounds, respectively. That is, the sum of the tensions will be 200 pounds, whether the belt is at rest or in motion. Now, suppose that the effect of gravity is to increase the tensions when at rest to 110 pounds, making their sum 220 pounds. When the belt starts both sides will still have the added 10 pounds tension, or the tension in each will be 135 and 85 pounds, respectively. Here we have $135 + 85 = 220$, as at the start the sum of the tensions not having changed.

The second point where we do not agree, and the last that I will take the room to mention, is where the statement is made that I indiscriminately attach too much importance to the arc of contact. To this I can simply say that the authority for my argument on this point is any reputable book in mechanics that my critic may choose to purchase. They were not original with me, although the method of treatment was entirely so. If I made no mistake in calculation, the figures which I give will be found to agree with those of any standard authority when reduced to his terms. If I am wrong, therefore, it is because the commonly accepted theory of belting is wrong, and the blame rests with me only in that I have examined the results of many belting tests made by different people, and feel very well assured in my own mind that the common theory of belts is nearly enough correct for all practical purposes. That was the reason why I gave the figures that I did.

Furthermore, I must pronounce Mr. Dunbars method of taking account of the arc of contact of the body on the pulley as totally wrong in every respect. The friction of a belt on its pulley at any point is dependent upon the normal pressure upon the pulley at that point. The case is similar to that of the pressure of water upon the sides of a pipe, and it would most certainly require a liberal amount of sleight of hand performing to produce correct results by the method proposed. The Editor would probably not print the mathematical reasoning if I attempted to give it, and so I must ask the readers to look the matter up for themselves. I am willing to abide by the verdict, whomever it hits. RETSEL.

NOTE.

Editor MACHINERY: In reply to writer on page 26, September, 1898, issue, would say that I was draughting for the Wm. A. Harris Steam Engine Co., Providence, R. I., making drawings of a governor, and thought out the idea of the overbalancing cam, and worked it up at home, and further, that it was built and used by that concern. WM. F. TORREY.

Roxbury, Mass.

[The foregoing is in reply to the letter of explanation in the last issue by Mr. Richard H. Rice, of the Rice & Sargent Engine Co., regarding the origin of a governor stop motion described by Mr. Torrey in a previous issue, and the patents upon the same.—Editor.]

SOLUTIONS TO MR. ROGERS' CHUCK PROBLEM.

The problem suggested by Mr. Rogers in your September issue interested me somewhat, as I studied out a formula for the angle of the jaws when the chuck first came under my notice a number of years ago. Mr. Rogers proposes to make the V in the chuck jaw a right angle, and then get an approximation of the thread required on the screw, with the strong probability of not having the gears to cut this screw. As I can see no reason for the V being an exact right angle, it seems to me an easier and more practical way out of it would be to assume for the screws certain pitches that could readily be cut, and then make the angle of the V to suit. To solve arithmetically, as Mr. Rogers suggests, take the reciprocal of the square root of 2, and multiply it by 10, which will give 7.0711 threads per inch. This does not look very inviting to a lathe hand, but if we assume the threads to be 7 and 10 per inch, respectively, and divide 7 by 10, we find on looking for the result in a table of sines that it belongs to an angle of 40° 26′, which is half the angle of the V; 44½° would be near enough, and the man on the shaper could set the head over to this angle almost as easily as he could to 45°. WALTER GRIBBEN.

Brooklyn, N. Y.

NUMBER TWO.

The following is offered in solution of the chuck problem propounded by Fred E. Rogers:

Assume two positions of jaws *A* and *B*, first as in Fig. 1, touching each other and next in Fig. 2, moved apart far enough to grasp a piece 2 inches in diameter.

Jaw *B* will then have moved back 1 inch or half the diameter

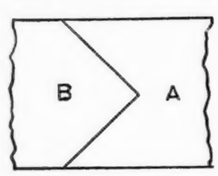


FIG. 1.

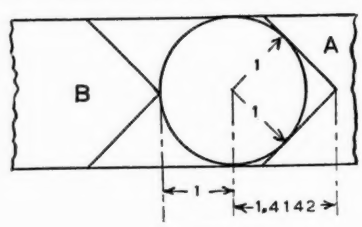


FIG. 2.

of the piece held, and the screw moving the jaw has made 10 turns to give this movement. Jaw *A* in the same time moves through a distance equal to the diagonal of a square whose side is 1 inch.

We know that the square of one side plus the square of the other equals the square of the hypothenuse, and we find that the square root of 2 is 1.4142, the distance moved by *A*.

As *B* moves 1-10 inch with each turn of screw, *A* will move 1-10 of 1.4142, or .14142. Diving 1 inch by .14142 gives 7.0711 as the pitch of screw for *A*.

In cutting 10 pitch on the latter with 4 pitch lead screw, which seems coarse, by the way, for a suitable sized lathe for this job, the change gears used would probably be 24 and 60. Replacing them by 56 on stud and 99 on the screw would give the above fractional pitch extremely accurately.

Or using the 24 and 60 change gears and introducing a compound gear between them made up of 70 and 99 running together, as one 24 on the stud running with 70, and 60 on the

screw meshing with 99, will give very accurate pitch. With this last arrangement the lathe may be set up with regular change gears for any pitch of screw desirable for *B*, and then by introducing the compound gear will cut the proper pitch for the jaw *A*, no matter what pitch had been chosen for *B*, or what the pitch of lathe lead screw is. ROBT. S. BROWN.

New Britain, Conn.

NUMBER THREE.

Referring to an article under the title of "A Chuck Problem," by Fred E. Rogers, published in the September, 1898, number of MACHINERY, would say if the salient and re-entering angles of the chuck jaws are 90 degrees, and the end of the screw which engages the jaw *B* is 10 threads per inch, the end which engages the jaw *A* should be 7.071068 threads per inch. This thread will be produced very closely in a lathe having a lead screw of 4 threads per inch, by using as drivers gears having 47 and 47 teeth and driven gears having 55 and 71 teeth.

Providence, R. I. R. T. WINGO.

APPRECIATED.

Editor MACHINERY: Permit me to thank you for the article in your September number, the "Doings of the Shellfish Club."

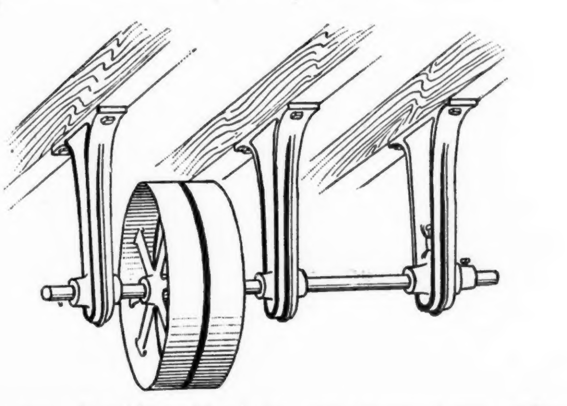
Some publishers have printed pages upon the cost subject, which have been a source of amusement to many people in the machinery business, who, while having long experience, still expect to learn more. But the elementary ideas put forth on the selling price and cost system with the indorsement of high priced journals has been something more than such people can understand.

I have no doubt that there are several hundred people in the machinery business who, if they were to speak their mind, would say that your treatment of the case has been very satisfactory, and will be the means of very delicately informing the would-be authorities upon this subject that they know nothing about which they write. Yours truly, COST KEEPER.

* * *

LOOSE PULLEYS.

The following is by a contributor to "The Woodworker": "This is the way I manage loose pulleys—or, rather, how I manage without them: Put up a short piece of shaft and let the end butt close to the countershaft. Put a pulley on the end so as to come close to the pulley on the counter, leaving them just far enough apart so they will not rub against each other, say ¼ inch; this will allow the belt to shift from one to the other. Both the pulleys thus being tight, when the belt is shifted on what is generally the loose pulley, the shaft turns instead of the pulley on the shaft. Of course this will not take the place of the loose pulley on the machine, but most of machines are supplied with patent loose pulleys, which give no trouble when properly taken care of.



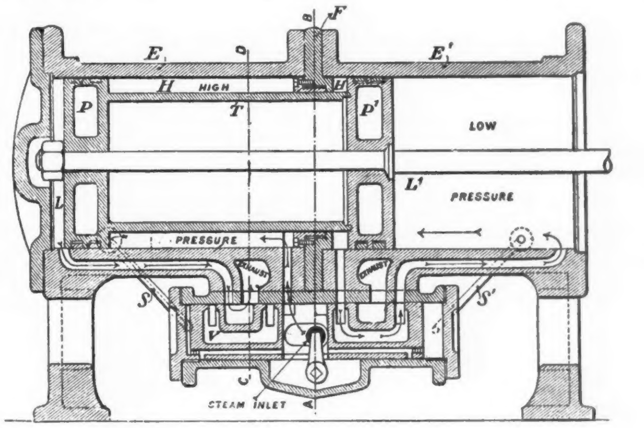
"This way of getting rid of the troublesome loose pulleys is the easiest, cheapest and best I know of. If any one else has a better way, I would like to learn it. I never have any trouble with loose pulleys."

This idea is an old one, but it is nevertheless a good one. In machine shop work there are comparatively few places where it could be applied because of the difficulty of arrangement, where the room is limited, and it probably would not find great favor in any case on account of the expense. There has been no end of brain power expended devising self-oiling loose pulleys and loose pulleys that can be oiled without shipping the belt onto the tight pulley and starting the machine. Of this latter class the simplest that we have seen is used on F. E. Reed's countershafts. The loose pulley is held in place on the shaft next to the tight pulley by a collar fast to the shaft, in the usual way. Then, instead of drilling the oil hole in the hub of the loose pulley, it is drilled in the collar, and a groove cut in the shaft extends underneath the hub of the loose pulley, from the bottom of the oil hole. The loose pulley can therefore be easily oiled while it is running, since the collar containing the oil hole is not then turning.

* * *

NOVEL COMPOUND PUMP.

A novel pump was recently illustrated in London "Engineering," made by Messrs. John Pickles & Son, of Hebden Bridge, England. A section of the compound steam cylinder is shown in the figure, and it will be noticed that it is divided by a central partition ring *F*. The trunk *T* which works through the partition *F* has a piston *P* or P^1 upon each end. The annular spaces *H* and H^1 outside the trunk receive the high pressure steam alternately, while the full area of the cylinder at the outer ends of the pistons *L* and L^1 forms the low pressure cylinder. The steam chest, which is bolted under the cylinder, is fitted with a piston valve *V*, which, again, is actuated by live steam conveyed by the pipes *S* and S^1 from the high pressure areas *H*. The live steam



enters the center of the piston valve *V*, and flows through short steam ports, in the direction shown by the arrows, alternately to the high pressure areas of the cylinder *H*. The pistons then make a stroke and cause a reversal of the piston valve, which then allows the high pressure steam to be conveyed to the low pressure area, and, by another stroke of the piston valve, thence to the exhaust. The location of the steam chest allows all condensation to drain to the exhaust; therefore drain cocks are not required. It will be noticed that the piston rod has but one packing gland, which is exposed to low pressure steam only. The packing ring, which clips the trunk, and is contained in the central partition ring *F*, is a special feature in this compound arrangement, and may easily be renewed by withdrawing the trunk and removing the screws in the junk ring.

* * *

One of the boys was busying himself by covering the edge of a large steam engine piston standing on the shop floor with a good, rich lather of B. T. Babbit's Best soap, using a long-haired paint brush for the purpose. He wasn't going into the barber business, as might have been inferred from the different questions that were put to him while he was at work. He was adopting this method of finding whether the piston was tight or not. The piston consisted of a light steel casting for the body, with two pieces of boiler iron riveted on to form the sides. It was constructed after this design in order to secure the greatest possible lightness, and thus bring the weight of the reciprocating parts of the engine to the lowest possible notch. The designer wanted to make sure that the piston was tight so that it would not fill up with water after the engine was started, but the webs in the steel casting were so placed that if the piston had been tested under water pressure it would not have been possible to have drained out all the water. Air pressure was therefore resorted to, obtained from the blower main in the foundry. It was first determined to detect the escape of the air by holding lighted matches near the surface of the metal, with the idea that the escaping air would blow the flame. The better plan was hit upon, however, of covering the surface with soap suds and letting the air blow soap bubbles wherever there was a blow-hole and the air escaped.

THE LEVER.—1.

MOMENTS AND THE PRINCIPLE OF MOMENTS AS APPLIED TO THE LEVER.

The lever is the simplest element of a machine, and the principles of its action are of a simple nature. There is no reason why any one who chooses to devote a little time to study should not be able to master these principles, and having done this, he will have gone a long way toward mastering the principles of all the elements that make up a machine.

Webster defines a lever as "a bar of metal, wood or other substance, used to exert a pressure or to sustain a weight, at one point at its length, by receiving a force or power at a second, and turning at a third on a fixed point called a fulcrum. It is of three kinds, according as either the fulcrum *F*, the weight *W* or the power *P*, respectively, is situated between the other two."

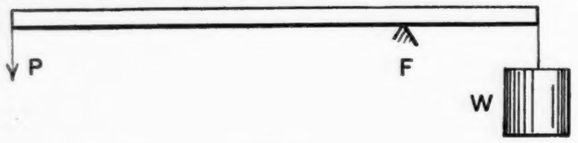


FIG. 1.

This is the usual definition of a lever as it is found in most books upon mechanics and physics, and attention should be called to certain points about it that could easily lead a beginner astray and cause confusion at the outset. It is always best to start with a clear idea of a subject, so that there will be no uncertainty to begin with, however much there may be later.

In Fig. 1 is a lever, in which, according to the definition, *W* is a weight acting at one point, *P* is the power or force acting at another point to raise the weight *W*, as indicated by the arrow, and *F* is the fulcrum on which the lever turns. That part of the lev-

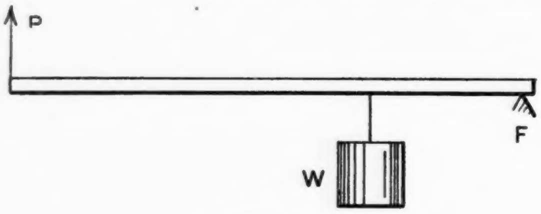


FIG. 2.

er between the weight and the fulcrum is called the "weight arm," and that part between the fulcrum and the power is called the "power arm." It will be noted that the fulcrum in Fig. 1 is located between the weight and power. In Figs. 2 and 3, however, are two levers in which the arrangement is different, the weight being placed between the power and fulcrum in Fig. 2, and the power placed between the weight and fulcrum in Fig. 3. These three figures illustrate the first, second and third kinds of lever, as above defined.

The objections to this definition of the lever are, in the first place, the use of the word "power" for the force applied at the end of the lever to raise the weight. "Power" has a totally dif-

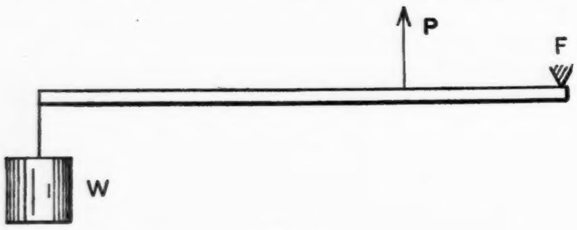


FIG. 3.

ferent meaning from "force," and takes into account not only force, but time and distance. A force is merely a push or a pull, such as is exercised by the hand, and this is the kind of effort that is always required to raise a weight or overcome any other resistance. In the reference letters of the illustrations, therefore, we will let *P* stand for a pull or a push, as the case may be, instead of for the word "power." Hereafter, also, in stead of calling the resistance to be overcome the "weight," we will call it the "resistance," and represent it by the letter *R*. A lever may have to overcome a number of resistances besides that of raising a weight, such as the resistance of friction, of a coiled spring or of the pressure of steam, and the term "resistance" implies this better than the term "weight."

Finally, regarding the three kinds of levers mentioned above, there is no necessity for trying to separate levers into any number of classes or for trying to remember to which class they belong in the solution of examples. All levers depend upon the same principles, which are simple and easily understood, and all that is necessary is to first master these principles without regard to the relative position of the applied force, the resistance or the fulcrum.

The Moment of a Force.

We have seen what is meant by the term "force," and the next thing to learn is what the moment of a force is. When a force acts at a point on a lever, that is, when that point is given a push or a pull, the tendency is to cause the lever to turn about its fulcrum. This tendency depends first upon the strength of the force acting and second upon the perpendicular distance from the line of action of the force of the fulcrum. If either the strength of the push or pull exerted by the force, or the perpendicular distance of its line of action from the fulcrum, is changed, the tendency of the force to rotate the lever will be greater or less, as the case may be. The rotative effect of any force thus depends upon both the strength and the distance, and is measured by their product, this product being called the moment of the force.

The moment of a force, therefore, is the measure of the turning effect of that force, and is found by multiplying the force by the perpendicular distance from its line of action to the fulcrum. If the force be measured in pounds and the distance in feet, the moment will be in foot-pounds; if the force be in pounds and the distance in inches, the moment will be in inch-pounds; if the force be in tons and the distance in feet, the moment will be in foot-tons, etc. The foot-pounds measurement is the most used, however.

This subject of moments is important—in fact, the most important in the whole subject of levers—and in order to fix it firmly in the mind, it will be helpful to have some common fact or operation that will illustrate it and that can be referred to in solving complicated examples in which the application of the principle may not be entirely clear.

There is one kind of lever that is very familiar to every mechanic, and that is the wrench. We will select the wrench, therefore, to illustrate the subject of moments, and having once grasped the principle as applied to the wrench, no mechanic will be likely to have trouble with its other applications.

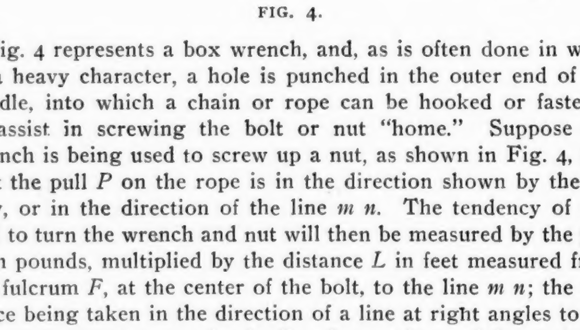


FIG. 4.

Fig. 4 represents a box wrench, and, as is often done in work of a heavy character, a hole is punched in the outer end of the handle, into which a chain or rope can be hooked or fastened to assist in screwing the bolt or nut "home." Suppose the wrench is being used to screw up a nut, as shown in Fig. 4, and that the pull *P* on the rope is in the direction shown by the arrow, or in the direction of the line *m n*. The tendency of this pull to turn the wrench and nut will then be measured by the pull *P* in pounds, multiplied by the distance *L* in feet measured from the fulcrum *F*, at the center of the bolt, to the line *m n*; the distance being taken in the direction of a line at right angles to the line *m n*. In other words, the line *L* must be at right angles or perpendicular to the line *m n*. This product gives the effect of the pull *P* in foot-pounds, and is called the moment of this force. Thus, if the pull *P* is 300 pounds, and the length *L* is 4 feet, the moment of the force *P* is $300 \times 4 = 1{,}200$ foot-pounds, and this is the measure of the turning effect of this force.

The reason why this is so will be evident if we consider another case shown in Fig. 5. Here the wrench has been placed in a new position, ready for another turn, and the pull *P* acts in the same direction as before, along the line *m n*. Now, anybody

who has used a wrench knows that with the same pull a greater effect will be produced with the wrench as placed in Fig. 4 than as placed in Fig. 5, although in each case the hook is at the same distance (4 feet) from the fulcrum *F*. The direct distance, however, of the point of application of the force from the fulcrum does not necessarily have any influence on the effectiveness of this force in moving the lever. The only distance that can be considered is the perpendicular distance from the line along which the force acts to the fulcrum, and this distance is greater in Fig. 4 than in Fig. 5, and in the former the force of 300 pounds has a greater leverage than in the latter. In Fig. 5 the measure of the rotative effect is the pull *P*, which is 300 pounds times the distance *L*, which in this case measures 2 feet, or $300 \times 2 = 600$ foot-pounds The distance *L*, as before, is measured at right angles to the line *m n*, and if the rope had extended along the line *c d*, instead of the line *m n*, *L* would have been taken at right angles to the line *c d*, as indicated by the line L^1.

The True Lever Arm.

The distance *L* in Figs. 4 and 5 is called the lever arm. Ordinarily the arm of a lever is understood to mean that part of the lever that lies between the fulcrum and the point where the force is applied, or between the fulcrum and the point where the resistance takes place; and such it is in a strict sense if the lever arm is straight and the force acts at right angles to the lever. But in Fig. 5 the true length of the lever arm is the distance *L* and not the length of the handle of the wrench, because *L* is the effective length acting in the position shown.

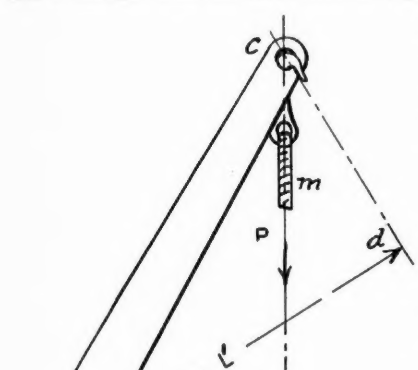


FIG. 5.

The true lever arm, therefore, is the perpendicular distance from the line of action of the force to the fulcrum.

A familiar example of the moment of a force is to be had in the action of the foot in pedaling a bicycle. When the crank has passed the upper center and the foot is ready for the downward push, it will require a much greater effort to drive the wheel ahead than when the crank is at right angles to the direction of the motion of the foot. The crank, of course, is of the same length whatever its position; but considered as a lever, the length of its arm varies from nothing at the upper center to the full length of the crank at the extreme forward movement of the foot. The moment of the force exerted by the foot, therefore, gradually increases from nothing at the upper part of the stroke to the greatest amount at the forward position.

Still another illustration is to be had in the curved crank shown in Fig. 6. The crank turns about the point *F*, and a rod is attached at the outer end which pushes in the direction shown by the arrow. Drawing the dotted line *m n* through the point at which the push is applied and in the direction in which the push is exerted, we have *L*, which is drawn at right angles to *m n*, as the length of the lever arm and the moment of the force is the length *L* multiplied by the force *P*.

The Principle of Moments.

Thus far the illustrations that have been used have pertained to what might be called single-armed levers. We have considered only the forces acting without regard to the resistances that had to be overcome, and the levers themselves have been more of the nature of a crank than of a lever, though it is not always easy to draw a distinction between the two. It is evident, however, that wherever a force is exerted there must also be a resistance, as otherwise no initial force would be required to create motion. In the case of the wrench, the resistance was the friction between the threads of the bolt and nut acting at the end of a lever arm equal to the radius of the bolt; and in the case of the bicycle crank the resistance was at the rim of the bicycle wheel, the lever arm in this case being more complicated because of the sprockets and chain.

In Fig. 7 is shown a bell-crank lever pivoted at the fulcrum *F*. A pull *P* is exerted along the rod at the left, and this is balanced by another pull along the rod at the right, which acts as a resistance to the force *P*. To determine the relative rotative effects of the pull *P* and the resistance *R*, we must determine the moments of these two forces.

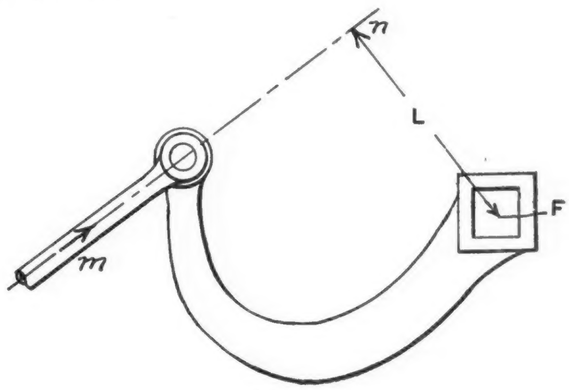


FIG. 6.

To find the moment of *P*, draw a line *m n* through the point in the lever at which *P* takes effect, and in the direction of the line in which it acts. Then draw the line *L* from the fulcrum *F* and at right angles to the line *m n*. This will be the true lever arm, and the moment of *P* will be the product of *P* and the length *L*.

To find the moment of *R*, draw the line *c d* through the point of application of *R* and in the direction of *R*. Then draw the line *D* of a length equal to the perpendicular distance from *F* to line *c d*. This will be the true lever arm for *R* and the moment of *R* will be the product of *R* and the distance *D*.

Since the moment of *P* measures the rotative effect of this force and the moment of *R* measures the rotative effect of the resistance, it is clear that if the lever is to balance, these two moments must be equal. If *L* is longer than *D*, as it is in this case, then *R* must be enough greater than *P* to make up for this, or otherwise the lever would begin to turn about *F*. This in substance is all there is to the principle of moments. The principle states that, if a body is to be in equilibrium, the sum of the moments of the forces which tend to turn it in one direction

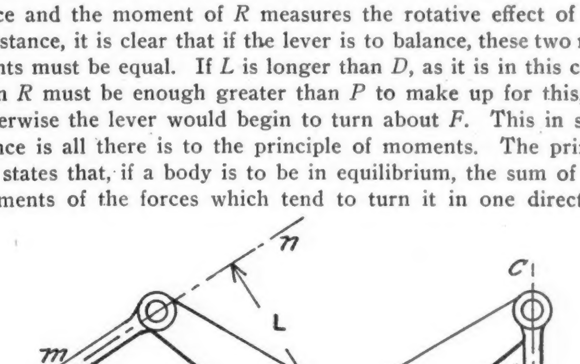


FIG. 7.

about a point is equal to the sum of the moments that tend to turn it in the opposite direction about the same point. In other words, if a body is to balance about a point, the opposing moments must be equal.

In the next article it will be shown how the principle of moments is applied to the solution of examples. F.

* * *

It is stated by the "Engineering News" that the largest dynamo in the world is now being constructed by the Walker Co., of Cleveland, O., for the Boston Elevated Railway, Boston, Mass. This generator will have an output of 3,000 kw. at 550 volts, or about 4,000 HP. Its speed will be between 75 and 80 revolutions per minute, its total weight 250,000 pounds, and the diameter of the circular cast-steel field frame 21 feet 7 inches. The weight of this ring without field magnets will be 25 tons. The armature hub is 13 feet in diameter and weighs 20 tons.

UNIVERSAL GRINDER.

The universal grinder, illustrated herewith, is the smallest size of the latest type of grinder made by the Diamond Machine Co., of Providence, R. I. It is designed to take work 18 inches long and 8 inches in diameter. As is usual with machines of this class, the travel of the table is automatic in both directions, and is controlled by dogs operating against a pin placed on the reversing lever. This pin can be dropped and the table moved beyond the reversing points without change of dogs. The table swivels on a central pin and the ends are finished on a radius from this and graduated to read to degrees. The adjustment is made by a tangent screw at the end of the table.

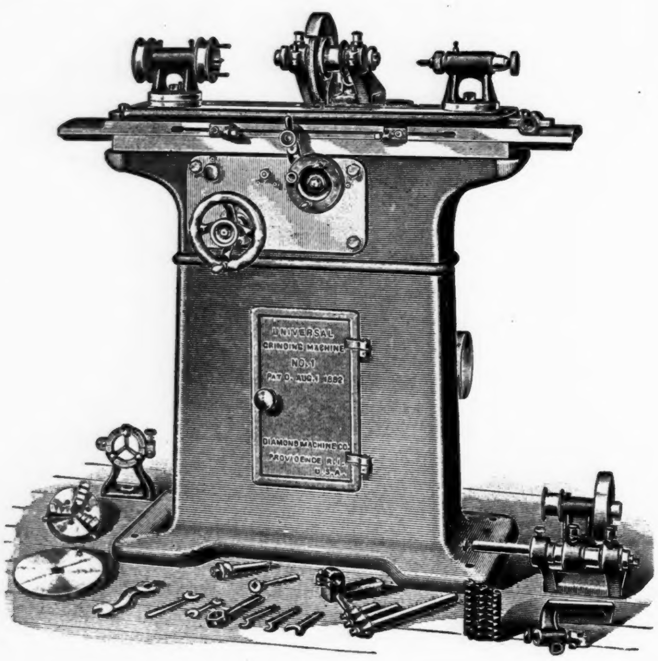


FIG. 1.

A section of the hand wheel used to adjust the emery wheel is shown in Fig. 2. By clamping the knurled nut on its face the stop pin may be set at any point and securely held, and the small finger or stop may be thrown back to allow the stop pin to pass, so that the wheel may be run back and brought up to the same place as before.

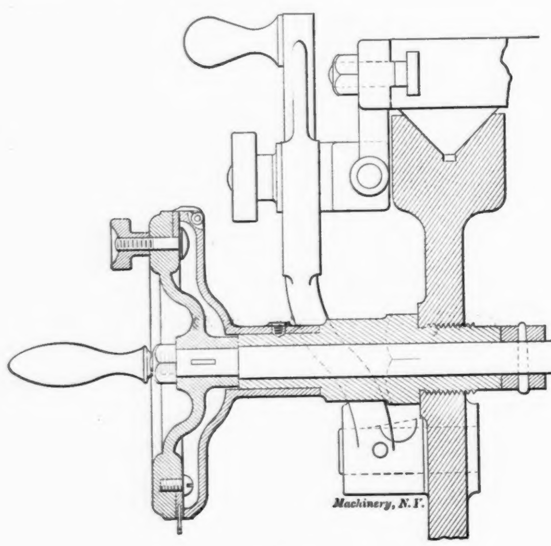


FIG. 2.

The foot stock spindle has a screw adjustment fitted with a spring, which adjusts itself independently to the varying length of work caused by expansion; or the spindle may be clamped by a knurled nut on the front end of the sleeve.

The internal grinding fixture and speed frame, the details of which are shown in Fig. 3, is independent of the emery wheel stand, and it is quickly placed on the machine and clamped by two bolts, the belts that drive being the same length on both heads. The spindle is of hardened steel accurately ground and runs in ball bearings, as shown, which are adjustable for wear. The tube carrying the outer bearing is clamped on the outer support, and the wheel arbor is fitted in the taper hole in end of spindle. These tubes have ball bearings near the emery wheel. They are easily removed, and can be of any length and diameter suited to the work. The belts are regulated by a swinging arm and thumb screw. Provision is made for excluding dust from the bearings and for holding the oil thrown from the spindle when running at a high rate of speed.

The wheel spindle is ground and lapped, and runs in bronze boxes provided with adjustments for wear. In the left hand end of the arbor is a hole which receives a taper shank for small emery wheels. For facing work the wheel can be removed by loosening four small screws from the flanges, and the arbor can be taken out and the emery wheel removed without disturbing the adjustment of the bearings.

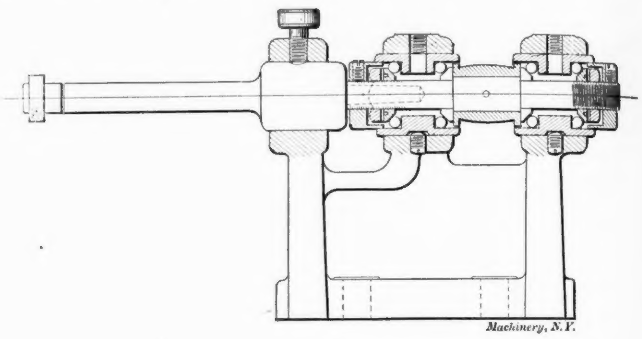


FIG. 3.

The head stock is attached to a base plate, which is bolted to the swivel table of the machine. It is made to swivel about a central stud, and its circumference is graduated to degrees and can be set at any angle. The spindle is provided with taper bearings, which are adjustable for wear, and has a lock pin for dead center grinding. The dead center pulley runs on taper bearings screwed on the nose of the spindle.

* * *

BOOK NOTICES.

Indicator Diagrams and Engine and Boiler Testing, by Chas. Day, Wh.Sc. Published by The Technical Publishing Co., Ltd., Manchester, England., and for sale by the D. Van Nostrand Co., Murray street, New York. Second Edition, 212 12mo pages, illustrated. Price, $2.00.

One-half of this work is devoted to the indicator and contains a very satisfactory treatment of the subject, even for an American engineer, who is accustomed to American methods. A large number of sample diagrams are given, showing defects, etc., and valve diagrams are taken up in this section. Several American indicators are also described, and their use explained. The second half of the work treats of engine testing. A considerable portion of this section is given up to the analysis of coals and temperature and analysis of flue gases, while a disproportionately small space appears to be devoted to certain other important subjects, like dynamometers, for example. The section on calorimetry is thoroughly up-to-date, treating almost entirely of the different forms of the throttling calorimeter. The observations that are to be taken in a test are explained, and directions given for making the calculations.

Painting to Prevent Corrosion, with Specifications. By A. H. Sabin, M.S. Published by Edward Smith & Co., 45 Broadway, New York. 84 pages, illustrated. Pocket size, bound in leather. Price, $1.25.

This is an excellent example of the class of literature which, while primarily intended to advance the interests of the company publishing the work, is of as much technical value as though it were a book issued through a regular publisher in the regular way. Edward Smith & Co. are manufacturers of paints for metal work, and they have done a good service for those who have structural work to be painted, or who have to write specifications for such work, in publishing the information that Mr. Sable has collected together in this book. There are two complete sets of specifications for painting structural work, accompanied by a full discussion of the various points, and the latter part of the book contains an article by Mr. Sabin, reprinted from the Troy "Polytechnic," giving much practical information.

HOW AND WHY.

A DEPARTMENT INTENDED TO CONTAIN CORRECT ANSWERS TO PRACTICAL QUESTIONS OF GENERAL INTEREST.

Give all details and name and address. The latter are for our own convenience and will not be published.

17. L. A. H. asks: How can I blacken pieces of sheet zinc so that it will retain the color and show fine lines traced on it for making templets, and will the same process apply to other metals? A. The following will give zinc a black color. On iron or steel the surface will be the color of copper: Add to 2 ounces of clean water ½ ounce of sulphate of copper and a few drops, about ¼ teaspoonful, of nitric acid. Clean the piece to be plated and rub it over with a cloth dampened in the solution and rub dry.

18. L. A. R. writes: We use a considerable number of friction clutches, and have a good deal of trouble with them. The clutches, various sizes of them, are made about as represented in Fig. 1. They work well at first, but gradually grow ineffective.

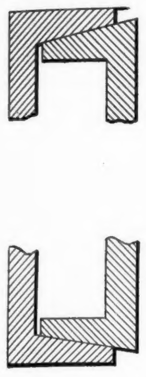


FIG. 1.

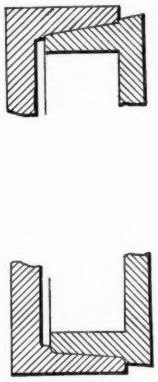


FIG. 2.

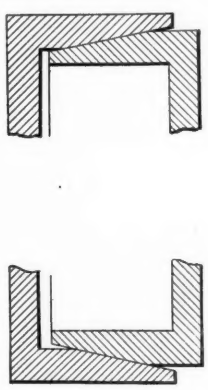


FIG. 3.

Can you suggest any reason for this? A. Clutches made like the sketch are rather certain to work, as exaggerated in Fig. 2. The remedy is to make them as represented in Fig. 3, there being then no chance to wear the shoulder, shown in Fig. 2.

19. T. L. M. writes: I have just started building Corliss engines, and would like to ask: What is the best way to proceed in setting the valves after the engine is erected? How are the knock-off cams set so as to cut off at 1-3, ½, ⅝, ¾ stroke, etc.? A. You cannot arrange the Corliss valve gear so as to release the steam valves later than ½ stroke, even if the valves have no lap. As it is important to give these valves a little lap, the unlocking must occur somewhere before half stroke. This you may readily see by considering that if there is neither lap nor lead when the crank is on the center, the eccentric will be at right angles to it, so that when the crank is set at half-stroke the eccentric will have moved the valve as far in the direction of opening as it will move it.

To overcome the lap and to give the valve lead, the eccentric must be advanced a little, and will be at its greatest throw ahead before the crank is at half stroke. The release must occur, if at all, before the valve has ceased moving ahead.

The first operation in valve setting (the engine is supposed to be horizontal) is to plumb the wrist plate; that is, to place the center of pin directly over the center of shaft and wedge it temporarily. By shortening or lengthening the connections bring both steam valves to the position of having the right lap, which for a cylinder less than 14 inches diameter may be ¼ inch, gradually increasing to ½ inch for a 36 inch cylinder; also adjust the connection so that the exhaust valves shall have from 1-32 inch to ⅛ inch lead for the size cylinders named. The wrist plate is now plumb, the steam valves are closed and ¼ inch more, and the exhaust valves are open by 1-32 inch (the cylinder is supposed to be 12 inches diameter). This part of the work should be carefully done, as the connections should not be changed in future operations of setting the valves. Next adjust the length of the eccentric rod, so that turning the eccentric around on the shaft it will move the carrier lever an equal distance each way from a plumb line. Then adjust the carrier rod so as to throw the wrist plate equally each way from the center. This is much easier done if marks have been previously established on pin and wrist plate. Place the engine on the dead center (either center) and move the eccentric around in the direction it is to revolve, until there is the proper lead—about 1-32 inch—on the end to which steam is to be admitted. Make the eccentric fast, go to the other center and observe if the lead is right there, which it will be if sufficient pains have been taken in all previous adjustments. The next operation is to adjust the unhooking gear. To do this block the governor up as far as it will go. Then adjust the length of the governor rods so that either end will unhook, the steam valve being open about 3-32 inch. Lower the governor to its lowest position, when steam should follow full stroke. It should follow full stroke until the governor balls have, in running, varied a considerable distance. When the engine is in operation apply the indicator and observe if the engine behaves satisfactorily. The governor rods may require a little further adjusting to suit the exact conditions under which the engine is being operated, but this will be a matter of small consequence.

20. M. A. J. writes: Is the dust from a large room full of woodworking machinery explosive? A. Explosion is no more nor less than rapid combustion. Therefore it is fair to conclude that the conditions may be such as to bring about an explosion of wood dust, although the proper conditions are not very likely to occur.

21. J. E. F. writes: I sometimes have tubes—copper, brass and iron—to bend, and I find difficulty in keeping them round. Can you tell me how they may be bent and kept in shape? A. Tubes that do not have to be heated for bending may be filled by melting rosin and pouring into them. This stiffens the tube while bending, or, in other words, since the pipe when round will hold the most, the tendency is to keep round. When the pipe is bent to shape heating the pipe will melt the rosin. Pipes that must be heated to bend may be filled with pure sand, well tamped down, the ends being then plugged.

22. S. A. asks: Will you give me a table for cutting the depth of gear teeth? I have a little gear cutting to do and always have to ask the depth to cut the tooth. A. The following is from the practice of Brown & Sharpe Mfg. Co, by permission. Dimensions are in inches:

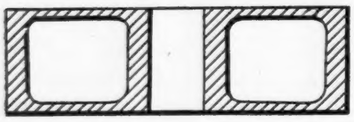
Circular Pitch.	Whole Depth of Tooth.	Circular Pitch.	Whole Depth of Tooth.	Circular Pitch.	Whole Depth of Tooth.	Circular Pitch.	Whole Depth of Tooth.
2	1,373	1¼	.858	¾	.515	⅜	.257
1⅞	1. 87	1 3/16	.816	11/16	.472	⅓	.229
1¾	1.202	1⅛	.772	⅔	.458	5/16	.215
1⅝	1.116	1 1/16	.729	⅝	.429	2/7	.196
1½	1.030	1	.687	9/16	.386	¼	.172
1 7/16	.987	15/16	.644	½	.343	2/9	.153
1⅜	.944	⅞	.6 1	7/16	.300	⅕	.137
1 5/16	.901	13/16	.558	⅖	.275	3/16	.120

To reduce circular pitch to diametral pitch, divide 3.1416 by the given circular pitch.

23. R. E. J. writes: I am a machinist, and accustomed to making fits of from 2 inches to 10 inches diameter by the use of inside and outside calipers, and find no trouble in getting just about what I want. I am told that I should do better work and more of it by making the fit by micrometer measurement, making a definite allowance for inch diameter. It seems to me that I should never be able to do any better in this way. What are the advantages of one way over the others? A. If, as we infer, you are skilled in making fits in the way you speak of, you will not improve either quantity or quality by the change. On the contrary, you would probably do the best by calipering. If it were merely a question of turning pieces the exact diameter of a dozen holes, varying slightly in diameter, you could not hope to do more than approximately as close work by setting verniers as by calipering, but in making press fit there is a good deal left to judgment, and to judge correctly as to a loose or a tight fit between the calipers and piece being turned requires a good deal of practice. Similarly it requires a good deal of educating of the eye to measure close. You will undoubtedly do better by adhering to the plan with which you are familiar.

24. E. R. A. writes: I am using a boiler that is larger than is necessary for the work, burning a light fire. My friend tells me that I should burn less coal if I were to reduce the grate area. It will cost something to get new grates and arrange for their use, and I should like your opinion of it before going to the expense. Will less coal make the same quantity of steam with the smaller grate? My friend says that there should be 12½ pounds of coal burned per hour on every square foot of grate surface. A. You can only tell as to the saving of coal by trial. Your friend is probably about right as to the quantity of coal that should be burned for good results. But it is by no means an assured fact that you will save coal by using a smaller grate with the present boiler. It will depend something upon the present proportion of grate to heating surface, and perhaps upon other existing conditions, with which we are unacquainted. Before making any very extensive arrangements for using a smaller grate, we should reduce the effective area of the present grate by bricking off some of it at the back end. Lay two courses of brick there and chink it carefully with ashes and spread some ashes over the top of brick. This chinking and covering with ashes is to prevent any leakage of air. Try this for a week or two. If you know how much coal you are now burning you can tell if you save by reducing the grate area, and then make the grate area permanently smaller if the temporary stopping off does not suit your purpose; that is, if the smaller grate surface saves coal.

25. T. G. writes: I have pistons cast hollow, as in sketch, which I shrink on the piston rods. I first plug the holes for anchoring the core with screw plugs, then heat the piston and insert the rod. Generally I have good success, but occasionally a piston cracks, the crack invariably starting from one of the plugged holes, extending an inch or so. Why should not all the pistons crack? Why should the crack start from the plugs? Can I do anything to prevent cracking? A. To begin with, the hollow is full of air and plugged tight. Air, like any of the gases, expands by heat, and since the volume cannot increase there is an increase in pressure. If you were to screw the plugs tight enough you would break the plate, hence there is an intention to crack before the pressure inside is increased. The cast iron is weakened by heating. You will see that you have an excellent combination working to crack the piston. The remedy is to leave one of the plugs out till the piston is shrunk on the rod.



26. H. P. G. writes: I have a chimney, built two years ago. It is 50 feet in height. The intention at the time was, when occasion required, to build it higher. At present but one boiler is connected to the chimney, but I expect to put in another of the same size. If I increase the height of the chimney to 100 feet will it not do for the two boilers just as well as it does for one now? A. Very likely it will do for the two boilers, but not entirely, because its height is doubled. The draft of two chimneys of unequal height varies not as their heights, but as the square root of their heights. Thus the comparative draft of two chimneys one 50 and the other 100 feet in diameter is (about) as 7 is to 10. If, then, dependence were placed entirely upon increasing the height of the chimney, there might be a failure. But in the great majority of instances, notwithstanding figures are against it, boilers will generally do anywhere from 50 to 100 per cent. more work—more in quantity, not in intensity—than they are doing. In this view of the case the gain from increased height will probably make your chimney ample for the two boilers.

27. D. A. writes: I have been advised that by pretty thoroughly wetting the coal before putting in the furnace I shall save at least 10 per cent. of the fuel bill. If this is so, why? A. There is no saving, but, on the conrtrary, a loss from wetting lump coal. Sometimes when burning dust it is prevented from flying around and perhaps being carried over the bridge wall by wetting before charging, but even then is no fuel saved, or at least very seldom.

28. L. A. writes: We are troubled with bad water and our boiler gets very dirty—covered with scale. The boiler has always been tight until recently. I began using a compound that is highly recommended for removing scale, and the boiler has commenced to leak. I have stopped the use of the compound, as I think the leaks prove that it is eating awav the iron around the joints. Am I not right? A. Not necessarily nor probably. The compound is quite likely eating away the scale and deposit, leaving a minute passage for steam and water. Unless you know of the compound being successfully used in your locality you had better have it analyzed to determine if it is likely to attack the metal of the boiler. If your boiler continues to leak, have it examined by a competent boiler maker.

29. A. K. asks: (1) If I set my boiler—a return tubular—so as to return the hot gases over the top of the shell, will there not be a saving of fuel? A. That will depend a good deal upon the length of boiler and diameter of tubes. If your boiler is short and the tubes fairly large, you would probably effect a little saving, but you might endanger the boiler. This—endangering the boiler—would depend, mainly, upon the quality of the coal used. If it contained any considerable amount of sulphur it would probably attack the sheets to some extent. We should not try setting this way, although we have seen boilers set so they have been in operation for ten years or more without the least apparent injury; on the other hand, we have heard of serious injury from such setting. (2) I propose to make an arch of 10 inches over the boiler through which to return the gases. Would this be about right? A. We cannot say, because we know nothing of the construction of your boiler. Make the arch of sufficient height so that at its point of greatest contraction, which will be at the dome, if there is a dome, the area of the passage will be 20 per cent. greater than the combined area of the inside of tubes. We say again that we do not advise such setting.

30. J. H. S. asks: In cutting pipe threads on pipe and in flanges over 6 inches in diameter, what is the proper way to set threading tools. The practice in our shop is to cut the pipe taper with the compound rest and set the tool to the taper, but the flange is bored straight and the tool is set square with the bore, and we get the taper by feeding our threading tool out by hand, as the tool feeds in. Please tell me the proper way, as all sizes over 6 inches are done on a lathe. A. The size of the flange is immaterial. Set the thread tool square with the axis of the flange, or precisely as if you were about to cut a piece bored straight. Your practice is wrong.

31. J. C. R. writes: If I have an engine all set in place and ready to run, how can I tell in which direction it is to turn over? A. Place the engine on the dead center. If the valves are set properly it will then have lead, no matter which way it is to run. Bar it a little one way and then the other. When you bar it the way it is to run, the valve will continue to open the port for the admission of steam.

32. M. W. writes: I have built a small haulage engine. The engine has four driving wheels, but the wheels are not coupled, and the engine does not haul so well as I expected. Will it do better if I couple the wheels, and if so, how much better? A. That depends upon whether the single pair slip. If the cylinders are large enough to slip the single pair of wheels, you will gain by coupling. If the single pair do not slip, then you will lose a little.

* * *

FRESH FROM THE PRESS.

A Practical Treatise on Modern Gas and Oil Engines, by Frederick Grover, A. M. Inst. C. E. Published by the Technical Publishing Co., Ltd., Manchester, England, and for sale by the D. Van Nostrand Co., Murray street, New York. Second edition. 256 12mo pages, illustrated. Price, $2.00.

In the preparation of this volume the author's intention was to present matter that would be of material benefit to the designer of gas engines. Accordingly, the leading types of engines are described with some attention to their mechanism, and information is given regarding the arrangement of the engine room. This matter is followed by a chapter on gas engine design. Little or no space is devoted to thermodynamics, but a statement is made of the leading thermodynamic considerations that influence the design. A great deal of attention is given to testing and to gas engine tests, including a chapter on gas analysis. There are over 50 pages upon the petroleum engine.

Bicycle Repairing. A manual compiled from articles in "The Iron Age." By S. D. V. Burr. Fourth edition, revised and enlarged. Published by the David Williams Co., New York, 208 8vo pages, illustrated. Price, $1.

This volume is based upon a series of articles which were pub-

The term "cross-filing" applies to those filing operations in which the file is pushed endwise across the work. When in cross-filing the character of the work requires a heavy file, it should be held in both hands, as shown in Fig. 1, the end of the handle

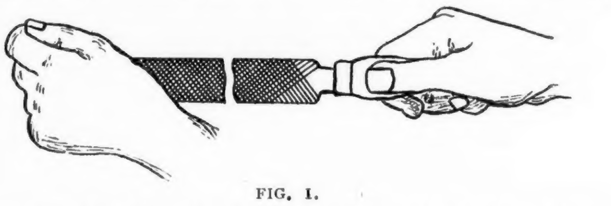


FIG. 1.

abutting against the palm of the hand, thus giving a good bearing to receive the thrust on the work stroke. When held in this manner an extremely tight grip is not required, which makes it much easier on the fingers and enables the workman to more readily control the file.

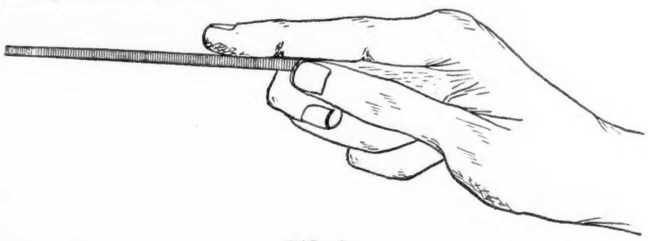


FIG. 2.

When a very light file is being used on fine work it is usually best to hold it in one hand, as shown in Fig. 2. In this case the thumb rests against the side of the file just ahead of the handle, and the free finger extends along the top, considerable downward pressure being exerted by this finger, as near as possible, over the working surface of the tool.

When the file is of medium size and thin, if held as shown in Fig. 1, the pressure at the ends will bend them down, making the file concave on its under surface, which will cause it to cut

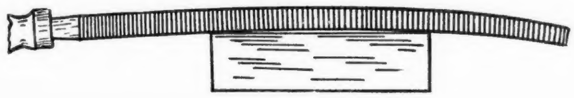


FIG. 3.

away the metal at the edges, as shown in Fig. 3. If, however, it is held as shown in Fig. 4, the downward pressure of the thumb will spring the file at the opposite direction, and thus enable the operator to move it across the work without cutting away the edges. When the thumb becomes tired, the position shown in Fig. 1 can again be taken, the ball of the thumb bearing down hard on the file and the fingers lifting at the point accomplishing the same object. Either of these methods of holding are difficult to maintain for more than a few moments at a time, consequently a stiffer file, having considerable belly, is preferable on work of this character.

The value of a good file handle should be appreciated. It should be of good size, well formed, smooth, properly ferruled and, most important of all, so secured to the tang that its center line is parallel with the length of the file. Handles made of soft, tough wood are preferable, as they are lighter and less liable to crack when forced on the tang. The soft wood handle, if provided with a hole for the reception of the tang of a diameter slightly greater than the thickness of the tang, can be driven

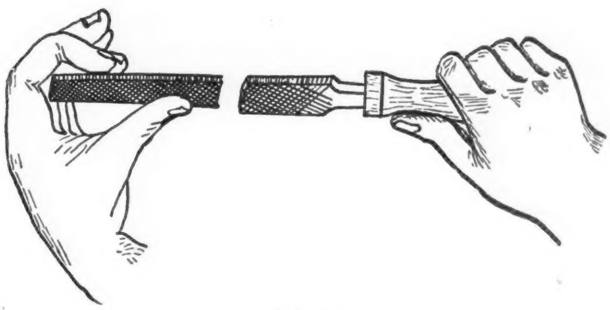


FIG. 4.

on without danger of cracking. If of hard wood a good job requires heating the tang red hot and burning the hole in the handle to fit it. Care must be exercised, or the temper of the teeth near the heel will be drawn. A piece of wet waste wrapped around the heel will prevent this.

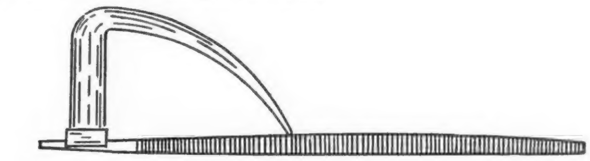


FIG. 5.

When the work surface is so broad that the file cannot be held, as shown in Fig. 1, on account of the handle striking against the edge of the work, a surface file holder must be used. In Fig. 5 is shown such a holder. The bottom of the handle is provided with a tapered, dove-tail slot to receive the tang, the outer point resting on the top of the file. Before applying the handle file the edges of the tang to approximately fit the dove-tail slot, as this may save a jammed set of knuckles. In using a file with this holder, the fingers of the left hand, resting on the top of the file, must give nearly all the pressure necessary to make it cut.

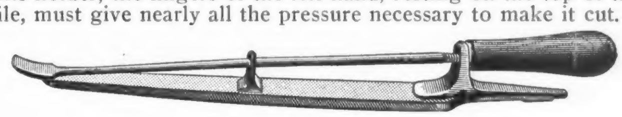


FIG. 6.

The form of surface file holder shown in Fig. 6 possesses the advantage of giving the operator a handle similar in shape and position to that used on ordinary narrow work. The rod enables the left hand to so grasp the point of the file that the downward pressure may be applied with less fatigue to the hand than in the case shown in Fig. 5. When the handle is screwed tight against the shoulder, the rod draws up on the point, thus tending to give the file more curvature, an advantage of considerable moment in filing accurate plane surfaces.

lished in "The Iron Age" in view of the need of information upon the methods of bicycle repairing. In collecting the material it has been the aim to confine the descriptions to such methods as have been tested and found practical in actual use. It is shown how each individual part of the wheel can be repaired, and in addition such subjects are included as the equipment of the shop, tempering and case hardening, enameling, nickel plating, keeping track of the work, etc. The illustrations are numerous, and many kinks are shown that will doubtless prove of value to any mechanic in that line of work.

AMERICAN STREET RAILWAY DIRECTORY AND BUYERS' MANUAL, Published quarterly by E. L. Powers, New York. Price, $3 a year.

This is the first number of the first volume of this publication. It contains a directory of the street railways of the United States and a buyer's finding list for buyers of street railway apparatus and equipment. There is also a page of newly compiled statistics of street railways of America. In the publication of this directory it is intended to make changes and corrections in each issue, so as to keep it up to date.

ADVERTISING LITERATURE.

THE STANDARD SIZES FOR CATALOGS ARE 9 x 12, 6x9 AND 3½x6 INCHES. THE 6x9 IS RECOMMENDED, AS THIS SIZE IS MOST LIKELY TO BE PRESERVED.

BUILDERS' IRON FOUNDRY, Providence, R. I. Catalogue of grinding and polishing machinery and patent countershafts. 112 pages, illustrated. 4½ by 6, standard size.

This catalogue illustrates and describes a full line of grinding and polishing machinery, and includes the "pull" countershafts made by this firm, which are already familiar to our readers. Among the new tools listed is a water tool grinder with a new device for carrying water to the wheel, which consists of a disk rotating in the water tank below the wheel. The disk throws up the water through the action of centrifugal force, thus insuring that the wheel will be well flooded when running, but will not stand in the water when not running.

BARDONS & OLIVER, Cleveland, O. Illustrated Price List of Improved Turret Machinery. 124 pages, 4 by 6 inches.

In this catalogue are listed the large line of turret machinery designed and constructed by this firm. There are nine distinct sizes of machines, ranging up to 20½ inches swing, and these are made in various styles, such as mounted on column with or without pan, mounted on legs, with different styles of heads and sizes of spindles and with numerous changes and attachments to adapt them to special work, such as bicycle hub forming, etc. These different types and attachments are illustrated in the catalogue and described very fully.

THE WILLIAM POWELL COMPANY, Spring Grove avenue, Cincinnati, O. Catalogue of high grade steam brass goods and specialties. 276 pages, illustrated, pocket size.

The goods listed include valves, lubricators whistles, boiler fittings, safety valves, injectors, etc., and all fittings of this class used about power plants and by steamship and railroad companies, etc. A number of pages are added containing tables and other useful information for pipe fitters and engineers that will be appreciated by those in this line of work.

THE DIXON CRUCIBLE COMPANY, of Jersey City, N. J., has published a little pamphlet upon brazing. It treats incidentally of brazing graphite, the application of which to bicycle tubes prevents the adherence of the spelter and so effects a saving in labor by making unnecessary the filling which is otherwise needful. The brazing crucible is described and instructions are given how to build and set the furnace.

THE NEW HAVEN MANUFACTURING COMPANY, New Haven, Conn., have issued a circular of lathes, planers, slotters and drill presses. The lathes are from 18 inches to 75 inches swing, the planers take work from 20 to 100 inches in width, and the drilling machines of from 28 to 40 inches swing. They are all of the well known types made by this company.

We have received from the Rohn Roper Bearing Works, of Wolverhampton, England, a pamphlet containing a number of half-tone views of the different departments of these works. We note a liberal sprinkling of American tools among the equipment.

THE NICHOLSON FILE WORKS, of Providence, R. I., have issued a wall hanger illustrating a new horse rasp that they are now supplying to the trade. These rasps are made of high carbon steel, thoroughly tempered.

"IF GOLD RUST, WHAT WOULD IRON DO?" is the title of a pertinent little circular issued by the Meriden Machine Tool Company, of Meriden, Conn.

MANUFACTURERS' NOTES.

THE LIDGERWOOD MANUFACTURING COMPANY, New York, a few months ago furnished a small traveling cableway for use in the construction of fortifications at Fort Pickens, Fla., the work being in charge of Major F. A. Mahan, Corps of Engineers, U. S. A. Lieutenant James P. Jervey, the engineer in local charge, has been able with this cableway to handle an average for fourteen days of 880 tons per day, the maximum day's work being 1,000 tons. The material was handled in skips of only 2 1-5 tons each, and this record made shows that there is frequently an advantage in running a light load fast instead of a large load slowly.

THE INTERNATIONAL CORRESPONDENCE SCHOOLS, of Scranton, Pa., have organized a correspondence course in telephony, the instruction papers for which were prepared by Mr. Kempster B. Miller, a well-known expert. The fact that it has been thought essential to supply a course in this one department of electrical engineering, apart from the other electrical courses, shows how thoroughly this enterprising institution is looking out for the interests of its present and future students.

THE LANE & BODLEY Company, engine builders, of Cincinnati, O., have called for bids for extensive new works, which will probably be erected at Norwood, adjacent to the new plant of the Bullock Electric Manufacturing Company. The large water works pumping engine contract which they have on hand necessitates greater facilities than the company now has, and there have been some heavy machine tools ordered for the work in hand, which will be the largest of their kind, and which will require a new establishment.

THE BAY STATE IRON WORKS, of Erie, Pa., manufacturers of engines, boilers and hoisting machinery, have opened a New York office at 141 Broadway, which will be in the charge of Woolston, Brew & Co., who have moved to New York from Pittsburg.

THE F. E. REED CO., of Worcester, Mass., report business as good, and state that they have numerous foreign orders, among them several heavy lathes for Russia.

MONTGOMERY & CO., 105 Fulton street, New York, offer their '98 tool catalogue during the months of October and November for 16 cents, postpaid. The usual price of the book is 25 cents, and it contains over 500 pages of matter descriptive of the large line of tools which they carry.

THE ALMY WATER-TUBE BOILER COMPANY, of Providence, R. I., supplied the boilers for the new police boat "Lexington," of Boston, Mass., which has made quite an unusual record for a boat of this class. It is only 122 feet over all, and in the trial run averaged 13.3 knots under natural draught.

Through an oversight in proofreading an error was allowed to creep into the advertisement of the Brainard Milling Machine Co., on page 26 of the September issue. The titles of the two lower illustrations were transposed.